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Solutions Manual for

Machine Elements in Mechanical Design, 5th ed.

By: Robert L. Mott

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MDESIGN Software - Its application to
Machine Elements in Mechanical Design, 5th edition
By: Robert L. Mott
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General Description of the Software

A powerful computer-aided calculation software package called MDESIGN is included with each purchase of this book. A total of 66 modules divided among 15 categories make up the complete package, outlined in the Introduction to the software. The software is an updated version of one that first appeared in the 4th edition of *Machine Elements in Mechanical Design*.

The software was created by TEDATA, GMBH, a German company that has a long history of producing such software for professional use throughout Europe and many other parts of the world. The version included with this book has 32 modules that have general applicability or that were produced specially for the book, following the analysis and design methods presented in the book, most of which are patterned on methods and standards commonly used in the United States. The other 34 modules were developed primarily for use by professionals and conform to common practice in Europe as represented by DIN standards, VDI publications, and the popular reference book on machine elements often called Roloff/Matek Machine Elements, written by Herbert Wittel, Dieter Muhs, Dieter Jannasch, and Joachim Vobiek and published by Vieweg+Teubner, Wiesbaden, Germany, 2009.

Identification of the two types of MDESIGN modules listed in the Introduction to the software:

- The list identifies the 32 modules most closely aligned with this book by the symbol ⊕. The chapter and section of the book most relevant to each module is indicated. The text includes several sections where a special icon appears to indicate that the use of MDESIGN is pertinent to that topic.
- The other 34 modules are denoted by the symbol ⊗ and are more closely aligned to European standards. They may use terminology, notations, and symbols unfamiliar to those experienced primarily in U.S.-based practices.

It is important to note that the inclusion of this extensive and diverse set of modules can be useful to users of this book throughout the world as a means of expanding the breadth of knowledge of design practices in different regions. Furthermore, many users of the book are likely to engage in projects with industrial companies, design services, consultants, and university faculty members from many parts of the world and having these modules available can aid in communicating across traditional geographic boundaries and between different technical cultures.

Advice on Use of the MDESIGN Software

The following comments are directed primarily to those using this book as a learning tool either in college and university degree programs or in professional self-study.

The author's approach to the inclusion of calculation aids within initial learning of technical subject matter is:

- *Users of computer software and calculation aids must have solid understanding of the relevant principles of design and stress analysis to ensure that design decisions are based on reliable foundations.*
- *Software should be used only after mastering a given design methodology by careful study and practicing manual techniques.*
- *Then, data with known results can be applied to the software as a check on the understanding of the program's input data requirements, symbols and notation used, limits on the range of acceptable data, and analysis methods.*
- *Only then should users rely on implementation of design decisions based on output results from the software.*

General HELP for Running MDESIGN

An extensive 123-page help file can be accessed from the main menu ribbon. Particular attention should be paid to the Graphical User Interface section on pages 33-39 for those few modules that permit graphical data input.

Recommended Primary Uses for MDESIGN Software with this Book

Upon launching the MDESIGN software package, reading the Introduction, and opening the software, the left side of the initial screen will include the list of 15 categories of modules. Each category name is preceded by a plus-sign (+) that, when selected, yields the list of modules in that category. Double-clicking will open any selected module. Alternatively, you can right click and select Open.

The 32 modules most closely aligned with the presentations in the book, identified by the symbol ⊕ in the Introduction, are obviously those that should be considered first for incorporation into courses and individual study. Pertinent sections of the book for which these modules may be useful are indicated by the graphic symbol in the left margin.

Particularly for design projects and where multiple trials for design decisions are to be expected and where the large catalogs of data in MDESIGN can be accessed, the following modules enable learners to try many options in a short amount of time after learning the basic fundamentals. The following modules serve well these purposes.

Beam Calculations	Column Analysis	ISO Fit System
Statically determinate beams	Column Design	Parallel Keys
Statically indeterminate beams	Ball and Roller Bearings	V-belts
Helical Compression Springs	Plain Surface/Journal Bearings	Synchronous Belts
Helical Extension Springs	Clutches and Brakes (5 modules)	Roller Chains
Helical Torsion Springs	Combined Stresses/Mohr's Circle	Shafts-U.S. Standards

Certainly, in an academic learning situation, instructors must enforce expectations on when and where use of the MDESIGN software is accepted, expected, or prohibited.

The process for using any module should be as follows:

1. Open the relevant module and read the General Text Help screen in the lower left part of the opening page. This outlines the basic functions of the module, shows the technical bases for the analyses performed, and identifies relevant references, terms, and symbols. Be aware that for some modules, the text help has been translated from the original German language and the result may not be in adequate standard English.
2. Use the pull-down menu on that screen to peruse what other textual aids are included. These often elaborate on design approaches by,
 - a. Explaining unfamiliar terms
 - b. Stating typical units for input data or results
 - c. Setting acceptable limits on values of certain variables
 - d. Providing tables of data from which some input data must be selected by the user
3. Observe the graphic aids in the lower right of the opening page, again scrolling among all available topics. Some of these can be accessed directly from the Input page.
4. Peruse the data required for the input screen. Open any available help icons for text, data, or “choice” options to determine requirements or available options.
5. Under the Tools tab on the main menu ribbon, select the pull down menu on the Measures System icon and select U.S. System, Metric System, or All Systems. These choices set the primary units in which data are to be entered and for which output results will be shown. In any case, you have the option to change units for any item by passing the cursor over the unit and pulling down the local menu.
 - a. Pay special attention to the precision of results data shown on the Output pages. At times, only one or two significant figures of accuracy are displayed and that may not be adequate for your use. You may be able to select a smaller unit that will show higher precision. For example if a length of diameter measurement shows 6.0 in, selecting the *mil* unit (0.001 – thousandths) may show 6075 mils, indicating 6.075 in.
 - b. Note that the standard European Metric system uses the comma rather than the decimal point for separating digits in floating point calculations. For example, In the U.S. system a number may be 12.456; in Metric it will appear 12,456.
6. When all data are entered, select the Calculation tab on the main menu or, simply, press the F10 key on the keyboard to initiate the module’s calculations. Note the following:
 - a. In some modules, intermediate data entry screens pop up for which some initial calculations have generated data on which subsequent design decisions are based. You are asked then to make the final decision before the complete results are found.

- b. After a short time for completing the calculations (typically only a few seconds), select the Output Page option at the upper left of the data page to see results.
 - c. Carefully evaluate results for reasonableness and check that proper units have been selected.
 - d. Some modules include internal checks on output to assess acceptability, with unacceptable results shown in RED. If that occurs, you must return to the Input Page and change design decisions, recalculate results, and re-evaluate their acceptability.
 - e. Consider the degree of optimization of any particular result and, where possible, make adjustments to hone into a more optimum design. *It is typical for mechanical design analyses to require proposing and analyzing several alternative solutions to achieve the most efficient and effective design.*
7. When the final result has been found, use the Print command to print out both the Input Page and the Output Page. It is essential for an instructor or a client to see complete records of the data used along with the results.

Descriptions of Selected Modules

The following sections describe certain topics from this book for which the use of MDESIGN is particularly pertinent. Suggestions for applying the modules are also given, but practice with known data is a good way to gain skill at entering data and seeking optimum results. Use of data taken from Example Problems from the text is highly recommended. However, there may be slight differences between results in book problems and those from MDESIGN because of rounding of numbers and slightly modified ways of making calculations.

Combined Stress and Mohr's Circle – Chapter 4

Module group: Shafts, Axles, and Beams. This module solves problems of the type featured in Sections 4-4 and 4-5 of the text in which data for applied normal and shear stresses in one plane are known and the program computes the maximum principal stress, the minimum principal stress, and the maximum shear stress. The complete Mohr's circle and the pertinent stress elements are also developed and included in the output.

Columns – Chapter 6 – Two modules: Column Analysis and Column Design

Module group: Shafts, Axles, and Beams. These modules follow closely the methodology used in the text for applying the Euler formula for long columns and the J. B. Johnson formula for short columns to either analysis or design problems. Loading can be either central or eccentric

and both straight and crooked columns can be analyzed. Problems of the types shown in the text in Example Problems 6-1 to 6-6

<i>Belt Drives and Chain Drives – Chapter 7 – Three modules: V-Belts, U.S. Standards, Synchronous Belts, and Roller Chains ISO 10823</i>

Module Group: Belt-, Chain Drives: These modules are pertinent to Sections 7-4 to 7-7. Each module contains large databases of commercially available products that can be selected for designs of power transmission drives. It is recommended that the use of this module be combined with student use of actual online catalogs of belts, sheaves, chains, and sprockets to specify part numbers and model numbers that can be specified for purchase.

- **Comments: V-Belts, U.S. Standards:** This module is pertinent to Section 7-4 in the text. Data entry and calculations are modeled after the method demonstrated in the book in Example Problem 7-1. Users are given options for selecting the belt size (3V, 5V, or 8V) and Figure 7-9 is used by the program to suggest a choice, which may be overridden by selecting another size. Selections for 'Driver' and 'Driven machine' types are identical to those used in Table 7-1 in the text. A design value for center distance is selected by the user after being given nominal minimum and maximum values. Then the user is presented with a set of optional combination of sheave sizes from which one must be chosen. That design is then evaluated and output data show the results. Iterations can be done easily by restarting the calculation and making modified selections.
- **Comments: Synchronous Belts:** This module approximates the methodology described in Section 7-5 of the text. Belts of the styles shown in Figure 7-18, both metric and U.S. sizes, are selected and analyzed by the program. Most data are shown in metric units for either style, although pull-down menus permit some features to be shown in U.S. units. The program is quite powerful, allowing multiple pulleys to be driven by one belt in serpentine arrangement. Most applications in this book will include two and only two pulleys. Data are input in tabular style and some practice may be required to become familiar with the details. It is recommended that the center of the driver pulley (No. 1) be positioned at $x = 0$, $y = 0$ on the coordinate system shown in the graphic aids. Then position the center of the driven pulley (No. 2; called a 'jockey pulley' in the module) at $x = \text{desired center distance}$ and $y = 0$. Entering 0° for the 'Displacement angle' will place the driven pulley to the right of the driver pulley. Enter 0 values for the 'max effective ϕ ' for each pulley and select 'within' for the 'Location' because the pulleys are positioned within the belt. For loads, it is normal to specify the power input to pulley 1 and the power output from pulley 2 to be equal and positive numbers. For U.S. data, select 'hp US' as the unit for power. Then leave the 'Torque' and 'Tangential force' entries as zero. The 'Load Factor' should be obtained from Table 7-1 (the same table as used for V-Belt drives). Then start the calculation. You will be presented screens offering choices for belt style, belt length, and belt width and you would normally select the nominal value offered for initial trials. Other values can be tried for subsequent trials until a satisfactory design is achieved.

- **Comments: Chain Drives – Roller Chains ISO 10823:** This module is pertinent to Section 7-7 of the text. It uses an ISO standard approach to the selection of chain drives that produces recommend chain sizes from ISO 606 as shown in Table 7-6 of the text. These sizes are identical to standard U.S. sizes for chain pitch as shown in the table. The performance analysis may differ slightly from the methods shown in the text. When using U.S. units for input data, ensure that Power is in ‘hp US’ and that units for other pertinent input and output data are expressed in the desired units, typically rpm for rotational speed and inches for dimensions. Selecting first the input page option: ‘Selection and calculation of one chain’ will result in a selection table being presented with options for different chain pitches and number of strands. Each design will list the rated power of the drive and the ‘Utilization’ (the value of the required corrected power to the rated power of the design, expressed as a percentage). The ‘Application factor to allow for the operating conditions, f_1 , is similar to the ‘Service factors’ shown in the book in Table 7-10. It is recommended that users select the option ‘Allow adjustment of factors f_1 and f_2 ’ (Yes). Then manually enter the service factor for f_1 and set $f_2 = 0$. This will match most closely to the methods used in the text. Common U.S. practice generally does not use a ‘Factor for number of teeth on drive sprocket’ (f_2).

Keys and Keyseats – Chapter 11

Module group: Shafts-Hub Connections: This module performs the calculations as described in Section 11-4 in the text, similar to Example Problem 11-1. Also included are calculations for the dimensioning variables Y , S , and T from Figure 11-2. Users enter the shaft diameter and either the torque of the combination of power and rotational speed. The yield strength of the key, shaft, and hub are entered or can be selected from a list of possible materials. After specifying the design factor to be used, the module produces the calculated results.

Rolling Contact Bearings – Chapter 14

Module group: Roller Bearings - Ball and Roller Bearings: The primary features of this module and its use are described in Section 14-11 of the text. The module provides access to a prominent manufacturer’s entire catalog for many types of bearings.

Plain Surface Bearings – Chapter 16

Module group: Journals – Plain Surface/Journals, US Standards: This module is patterned after Section 16-5 of the text – Design of Boundary-Lubricated Bearings. Problems of the types shown in Example Problems 16-1 and 16-2 are solved using this module. After entering data for radial load on the bearing, the rotational speed of the shaft, the selected trial value for shaft diameter, and a trial value of the L/D ratio, the program calculates the actual length, L , the bearing pressure, p , the surface speed, V , and the pV value, and the design value of pV .

(23calculated pV). It then searches a modest table of possible materials similar to that shown in Table 16-1 of the text for one that has a rated pV value closest to but more than the design pV value. The nominal diametral clearance for the bearing is also computed, using data shown in Figure 16-4 of the text.

- ***Please note that the label for the Diameter value on the Input page of this module uses the term, 'Nominal minimum diameter of the journal, D_{min} '. That is not the same as the minimum acceptable shaft diameter (based on shaft stress analysis). It should be the actual trial diameter for the shaft that is selected by the user and that must be graeater than the minimum acceptable value based on strength.***
- A recommended use for this module is as an aid in selecting commercially available sizes and materials for plain surface bearings from vendors such as those listed in the Internet sites for Chapter 16, particularly sites: 2 – Thomson Engineering & Polymers; 3 – Saint-Gobain Performance Plastics; 4 – GGB Bearing Technology; 5 – Graphite Metallizing Corporation; 6 – Beemer Precision, Inc.; and 7 – Bunting Bearings Corporation. These sites offer catalog data for their products and they list the design pV values for the various materials from which the bearings are made.
- For use of catalog data, users should select a preliminary value for the internal diameter and length for a particular bearing and make note of the design pV value for the selected material. Then compute the actual L/D ratio. Then enter the dimensions into the MDESIGN module (D and L/D), along with other given data for bearing load and rotational speed. The computed required "Design value of pV factor" should then be compared with the catalog-listed value for the selected material. The suggested material given on the module output page should be ignored. Iterations are easily and quickly done by trying other sizes until an optimum design is identified.

MACHINE ELEMENTS IN MECHANICAL DESIGN
Fifth Edition

Robert L. Mott

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Description of Spreadsheets Included with the Instructors Manual

Introduction

The Instructors Manual for this book contains a set of 26 computational aids that are keyed to the book. The files are written as Microsoft Excel spreadsheets.

Many of the spreadsheets appear in the text. Others were prepared to produce solutions for the Solutions Manual. The given spreadsheets include data and results from certain figures in the text, from certain example problems, or for certain problems from the end of chapters containing the analysis and design procedures featured in the programs.

The following sections give brief descriptions of each spreadsheet. Many are discussed in the text in more extensive detail. It is expected that you will verify all of the elements of each spreadsheet before using them for solutions to specific problems.

Using the Spreadsheets:

- ***It is recommended that you maintain the given spreadsheets as they initially appear on the disk, considering them to be master copies.***
- ***To use a program for solving other problems, call it up in Excel and use the "Save as" command to give it a different name.***
- ***For instance, the original program called Column Analysis should be considered the master. Use "Save as" and call it, for example, Column Analysis – Working. Then use that version for general problem solving.***

You should study the concepts and the solution techniques for each type of problem before using the spreadsheets. You should work sample problems by hand first. Then enter the appropriate data into the spreadsheet to verify the solution. In most spreadsheets in the text, the data that need to be entered are identified by gray-shaded areas and by italic type.

Descriptions of Spreadsheets

The descriptions are given here in the order that the subjects for the spreadsheets are covered in the text.

Column Analysis: Chapter 6. Analyzes straight columns of uniform cross section to determine the critical buckling load and the allowable load. The spreadsheet shows results for Example Problem 6-1. U.S. Customary units are used. A description is given in Section 6-8. The process is essentially the same as that shown in the flow chart of Figure 6-4. Note that a short macro program in Visual Basic is used to decide whether the column is *long* (Euler) or *short* (J. B. Johnson) and to complete the calculation of the critical buckling load. Be sure that your Excel program enables macros.

Column Analysis SI: Chapter 6. Same as **Column Analysis:** except SI units are used. The solution to Example Problem 6-2 is given as an example for data entry.

Circular Column Analysis: Chapter 6. Special version of **Column Analysis** in which the geometric properties of a column with a solid circular cross section are computed when the diameter is input. The spreadsheet can be used as an iterative design tool to determine the required diameter of a column with a circular cross section to carry a given load. See Figure 6-14.

Crooked Column Analysis: Chapter 6. Section 6-11. Analyzes the allowable load on a column of constant cross section with a given amount of crookedness. Data from Example Problem 6-4 are used as shown in Figure 6-16 on page 252.

Eccentric Column Analysis: Chapter 6. Section 6-12. Computes the required yield strength of the material and the resulting maximum deflection of the middle of a column that is loaded eccentrically. Data from Example Problem 6-6 are used as shown in Figure 6-18.

Chain Drive Design: Chapter 7. Design of roller chain drives as described in Section 7-6. User must obtain rated power data from Tables 7-7, 7-8, or 7-9 to specify a suitable chain number and number of teeth in the smaller sprocket. A service factor must be selected from Table 7-10 in the text. Data from Example Problem 7-4 are shown in the master spreadsheet.

Gear Geometry: Chapter 8. Computes the geometric features of spur and helical gears using the relationships in Sections 8-4 and 8-6. Can be used for Problems 1-9 and 41-44.

Contact Ratio-Spur Gears: Chapter 8. Computes the contact ratio for spur gears using the procedure shown in Section 8-4.

Bevel Gear Geometry: Chapter 8. Computes the geometric features of straight bevel gears using the formulas listed in Table 8-8 in Section 8-7 and illustrated in Example Problem 8-3. Two identical programs are shown side-by-side. One shows the results of Example Problem 8-3 and the other can be used to solve any given problem.

Wormgearing Geometry, C, VR: Chapter 8. Computes essential geometric features of a worm and wormgear, the center distance (C) between their shafts, and the velocity ratio, VR . Uses procedure from Section 8-9 as illustrated in Example Problem 8-4. The spreadsheet was used to complete Problems 52-57 at the end of the chapter.

Gears VR Design: Chapter 8. Aids in the specification of the number of teeth in a pinion and gear to produce a specified velocity ratio. Uses a procedure similar to that shown in Section 8-11 and illustrated in Table 8-10. An integer is entered for the number of teeth in the pinion. The program computes the required approximate number of teeth in the gear to produce the given velocity ratio. The user then enters an integer for the actual number of gear teeth. The program identifies the combination of numbers of teeth that produces the minimum differential between the desired ratio and the actual ratio. The spreadsheet was used to complete Problems 62-65 at the end of the chapter.

Spur Gear Forces: Chapter 9. Computes the tangential, radial, and normal forces on spur gear teeth of a given design transmitting a given power at a given pinion speed. It uses the method of Section 9-3. The spreadsheet was used to complete Problems 1-6 at the end of Chapter 9. The results for Problems 1 and 2 are shown in the master.

Spur Gears-Design-U.S.: Chapter 9. Performs a complete design analysis for a pair of spur gears, including the essential geometry, tangential force, required bending stress number, and required contact stress number. All modifying factors for stress calculations as described in Sections 9-8 to 9-11 are included. The data from Example Problem 9-4 are shown in the given spreadsheet as illustrated in Figure 9-25. An extensive discussion of the spreadsheet is given in Section 9-13. A feature of the spreadsheet is the computation of the required hardness (HB) for through-hardened Grade 1 steel using the equations in Figures 9-11 and 9-12. The user can then specify suitable materials and list them at the bottom of the spreadsheet.

Geometry Factor-I-Pitting: Chapter 9. Computes the value of the geometry factor, I , used in the calculation of contact stress for spur gears in Equation 9-23. Program uses the algorithm from Appendix A18.

Spur Gears-Design-U.S.-With I: Chapter 9. Same as **Spur Gears-Design** except the geometry factor, I , is computed within the program instead of being input by the user. The program **Geometry Factor-I-Pitting** is integrated within **Spur Gears-Design**. One additional input value is needed for the pressure angle, ϕ .

Spur Gears-Design-SI: Chapter 9. Similar to **Spur Gears-Design:** except SI metric data are used as described in Section 9-13 and illustrated in Example Problem 9-5. Data from Example Problem 9-5 are used in the given spreadsheet.

Spur Gears-Capacity-U.S.: Chapter 9. Section 9-15. Determines the power transmitting capacity of a given set of spur gears considering both bending strength and pitting resistance. The user must input the allowable bending stress and allowable contact stress based on the material specified for the pinion and the gear using Figures 9-11 and 9-12 and Table 9-5. The spreadsheet includes the computation of the required bending stress number, s_{ab} , and contact stress number, s_{ac} , based on user-entered hardness (HB) for through-hardened Grade 1 steel using the equations in Figures 9-11 and 9-12. The user must transcribe these values into the spreadsheet if, in fact, this kind of material is specified.

Plastic Gears- Design: Chapter 9. Completes the design of plastic gears using the procedure from Section 9-16. Data are shown for Example Problem 9-7.

Helical Gears-Design: Chapter 10. Computes the forces on helical gear teeth as described in Section 10-2 and illustrated in Example Problem 10-1. Completes the design analysis for a pair of helical gears as described in Sections 10-3 to 10-5 and illustrated in Example Problem 10-3. Used for the solutions to Problems 1-11 at the end of Chapter 10.

Helical Gears-Capacity: Chapter 10. Similar to **Spur Gears-Capacity:** with modifications for the special geometry of helical gear teeth. Used for the solutions to Problems 12 and 13 at the end of Chapter 10. The user must input the allowable bending stress and allowable contact stress based on the material specified for the pinion and the gear using Figures 9-11 and 9-12 and Table 9-5. The spreadsheet includes the computation of the required bending stress number, s_{ab} , and contact stress number, s_{ac} , based on user-entered hardness (HB) for through-hardened Grade 1 steel using the equations in Figures 9-11 and 9-12. The user must transcribe these values into the spreadsheet if, in fact, this kind of material is specified.

Bevel Gears – Design: Chapter 10. Computes forces and stresses on bevel gears using the methods shown in Section 10-9.

Wormgearing – Design: Computes worm and wormgear geometry values, forces, and stresses for wormgearing, using methods and data from Chapter 8 (Section 8-9) and Chapter 10 (Sections 10-10, 10-11, and 10-12). The master spreadsheet uses data from Example Problems 8-4, 10-9, and 10-10.

Keyseat Data: Chapter 11. Computes the data required to dimension keyseats and keyways on shaft drawings according to the information in Figure 11-2.

Shaft Design: Chapter 12. Computes the minimum acceptable diameter for shafts using Equation 12-24 when both bending and torsion are present and Equation 12-16 when only vertical shearing stress is present. Requires prior analysis for torques, forces, bending moments, pertinent material strengths, modifying factors on material strength, and stress concentration factor. The program is typically applied at several selected sections of the shaft as illustrated in Design Example 12-1 in Section 12-6. If the location being analyzed has a retaining ring installed, the computed minimum shaft diameter is considered to be for the base of the ring groove. The spreadsheet computes the nominal full shaft diameter by applying a factor of 1.06 as described at the end of Section 12-4. The data used in the master spreadsheet are for one location on the shaft in Design Example 12-1 as illustrated in Figure 12-19 in Section 12-9 where the spreadsheet and its use are described.

Force Fits: Chapter 13, Section 13-8. Stresses for Force Fits. Computes the pressure at the interface between mating members assembled with an interference fit (See Section 13-6.) Also computes the resulting stresses and deformations for the mating members using the procedure in Section 13-8. Data from Example Problem 13-2 are shown in the example.

Spring Design-Method 1: Chapter 18, Section 18-6. The given spreadsheet uses data and the method from Example Problem 18-2 to design a safe helical compression spring for a given loading and to fit given geometrical limitations. See Figure 18-16 and the accompanying discussion.

Spring Design-Method 2: Chapter 19. Similar to **Spring Design-Method 1** without the restriction of designing to a set of geometrical limitations. See Example Problem 18-3, Figure 18-17, and the accompanying discussion.

CHAPTER 1

THE NATURE OF MECHANICAL DESIGN

Problems 1 - 14 require the specification of functions and design requirements for design projects and have no unique solution.

15. $D = 1.75 \text{ in.} \times 25.4 \text{ mm/in} = \underline{44.5 \text{ mm}}$

16. $L = 46 \text{ FT} \times 0.3048 \text{ m/FT} = \underline{14.0 \text{ m}}$

17. $T = 12\,550 \text{ LB}\cdot\text{IN} \times 0.1130 \text{ N}\cdot\text{m/LB}\cdot\text{IN} = \underline{1418 \text{ N}\cdot\text{m}}$

18. $A = 4.12 \text{ in}^2 \times 645.2 \text{ mm}^2/\text{in}^2 = \underline{2658 \text{ mm}^2}$

19. $Z = 14.8 \text{ in}^3 \times 1.639 \times 10^4 \text{ mm}^3/\text{in}^3 = \underline{2.43 \times 10^5 \text{ mm}^3}$

20. $I = 88.0 \text{ in}^4 \times 4.162 \times 10^5 \text{ mm}^4/\text{in}^4 = \underline{3.66 \times 10^7 \text{ mm}^4}$

21. GIVEN $A_{\text{min}} = 750 \text{ mm}^2$; IN U.S. UNITS: $A_{\text{min}} = 1.162 \text{ in}^2$
 APP. 15-1: $L = 2 \times 2 \times 3/8$, $A = 1.36 \text{ in}^2 = 890 \text{ mm}^2$
 APP 15-3: ANGLES $50 \times 100 \times 6$ AND $75 \times 75 \times 5$ HAVE $A = 864 \text{ mm}^2$

22. $P = 7.5 \text{ hp} \times 745.7 \text{ W/hp} = 5.59 \times 10^3 \text{ W} = \underline{5.59 \text{ kW}}$

23. $S_u = 127 \text{ ksi} \times 6.895 \text{ MPa/ksi} = \underline{876 \text{ MPa}}$

24. LET $D = 0.035 \text{ m}$; $L = 0.675 \text{ m}$; VOLUME $= V = A \times L = \left(\frac{\pi D^2}{4}\right) \times L$
 $V = \frac{\pi (0.035 \text{ m})^2}{4} \times 0.675 \text{ m} = 6.49 \times 10^{-4} \text{ m}^3$

MASS $= \text{DENSITY} \times V = 7680 \text{ kg/m}^3 \times 6.49 \times 10^{-4} \text{ m}^3 = 4.98 \text{ kg}$

WEIGHT $= m \times g = 4.98 \text{ kg} \times 9.81 \text{ m/s}^2 = 48.9 \text{ kg}\cdot\text{m/s}^2 = \underline{48.9 \text{ N}}$

$$25. \quad T = 180 \text{ LB}\cdot\text{IN} \times 0.1130 \text{ N}\cdot\text{m} / \text{LB}\cdot\text{IN} = \underline{20.3 \text{ N}\cdot\text{m}}$$

$$\theta = 35^\circ \times \pi \text{ RAD} / 180^\circ = \underline{0.611 \text{ RAD.}}$$

$$\text{SCALE} = T / \theta = 180 \text{ LB}\cdot\text{IN} / 35^\circ = \underline{5.14 \text{ LB}\cdot\text{IN} / \text{DEGREE}}$$

$$\text{SCALE} = T / \theta = 20.3 \text{ N}\cdot\text{m} / 0.611 \text{ RAD.} = \underline{33.3 \text{ N}\cdot\text{m} / \text{RAD.}}$$

$$26. \quad \text{ENERGY} = \text{POWER} \times \text{TIME}$$

$$E = 12.5 \text{ hp} \times \frac{16 \text{ h}}{\text{DAY}} \times \frac{5 \text{ DAYS}}{\text{WEEK}} \times \frac{52 \text{ WKS}}{\text{YEAR}} \times \frac{550 \text{ FT}\cdot\text{LB}}{\text{S}\cdot\text{hp}} \times \frac{3600 \text{ S}}{\text{h}}$$

$$E = \underline{1.03 \times 10^8 \text{ FT}\cdot\text{LB} / \text{YEAR}}$$

$$E = 1.03 \times 10^8 \frac{\text{FT}\cdot\text{LB}}{\text{YEAR}} \times \frac{1.356 \text{ J}}{\text{FT}\cdot\text{LB}} \times \frac{1.0 \text{ N}\cdot\text{m}}{\text{J}} \times \frac{1.0 \text{ W}}{\text{N}\cdot\text{m} / \text{S}} \times \frac{1 \text{ h}}{3600 \text{ S}}$$

$$E = 38.8 \times 10^6 \text{ W}\cdot\text{h} / \text{YEAR} = \underline{38.8 \text{ MW}\cdot\text{h} / \text{YEAR}}$$

$$27. \quad \text{VISCOSITY } \mu = 3.75 \text{ REYN} \times \frac{1.0 \text{ LB}\cdot\text{S}}{\text{IN}^2\cdot\text{REYN}} \times \frac{144 \text{ IN}^2}{\text{FT}^2} = \underline{540 \frac{\text{LB}\cdot\text{S}}{\text{FT}^2}}$$

$$\mu = 3.75 \frac{\text{LB}\cdot\text{S}}{\text{IN}^2} \times \frac{4.448 \text{ N}}{\text{LB}} \times \frac{1.0 \text{ IN}^2}{645.2 \text{ mm}^2} \times \frac{10^6 \text{ mm}^2}{\text{m}^2} = \underline{25.9 \times 10^3 \frac{\text{N}\cdot\text{S}}{\text{m}^2}}$$

$$28. \quad \text{LIFE} = \frac{1750 \text{ REV}}{\text{MIN}} \times \frac{24 \text{ h}}{\text{DAY}} \times \frac{60 \text{ MIN.}}{\text{h}} \times \frac{365 \text{ DAYS}}{\text{YEAR}} \times 5 \text{ YEARS}$$

$$\text{LIFE} = \underline{4.60 \times 10^9 \text{ REVOLUTIONS}}$$

CHAPTER 2 MATERIALS IN MECHANICAL DESIGN

1. Ultimate tensile strength is the apparent stress at the peak of the stress-strain curve.
2. Yield point is the value of the apparent stress from the stress-strain curve at which there is a large increase in strain with no increase in stress. It is the point where the stress-strain curve exhibits a horizontal slope.
3. Yield strength is the apparent stress from the stress-strain curve at which there is a large increase in strain with little increase in stress for materials that do not exhibit a yield point. The offset method is used by drawing a line parallel to the straight part of the stress-strain curve through a value of 0.2% on the strain axis.
4. Many low alloy steels exhibit a yield point.
5. The proportional limit is the apparent stress on the stress-strain curve at which the curve deviates from a straight line. At this value, the material is usually still elastic. The elastic limit is the apparent stress at which the material is deformed plastically and will not return to its original size and shape.
6. Hooke's law applies to that portion of the stress-strain curve that is a straight line for which stress is proportional to strain.
7. The modulus of elasticity is a measure of the stiffness of a material.
8. The percent elongation is a measure of the ductility of a material.
9. The material is not ductile. Materials having a percent elongation greater than 5% are considered to be ductile.
10. Poisson's ratio is the ratio of the lateral strain in a material to the axial strain when subjected to a tensile load.
11. From Eq. 2-5 $G = E/[2(1+\nu)] = (114 \text{ GPa})/[2(1+0.33)]$
 $G = 42.9 \text{ GPa}$

12. Hardness = 52.8 HRC (Approximate; Appendix 17)
13. Tensile strength = 235 ksi (Approximate; Appendix 17)

14.-17. Errors in given statements:

14. A hardness of HB 750 is extremely hard, characteristic of the hardest steels in the as-quenched or surface hardened condition. Appendix 3 shows annealed steels to have hardness values in the approximate range of HB 120 to 230.
15. Hardness on the HRB scale is normally limited to HRB 100.
16. Hardness on the HRC scale is normally no lower than HRC 20.
17. The relationship between hardness and tensile strength is only valid for steels.
18. Charpy and Izod tests measure impact strength.
19. Iron and carbon. Other elements are often present.
20. In addition to iron and carbon SAE 4340 steel contains nickel, chromium, and molybdenum. (Table 2-8)
21. Approximately 0.40% carbon in SAE 4340 steel.
22. Low-carbon: Less than 0.3%
Medium-carbon: 0.30% to 0.50%
High-carbon: 0.50% to 0.95%
23. Typically a bearing steel contains 1.0% carbon.
24. Lead is added to SAE 12L13 steel to improve machinability.
25. Shafts are often made from SAE 1040, 4140, 4640, 5150, 6150, and 8650 steels. (Table 2-9)
26. Gears are often made from SAE 1040, 4140, 4340, 4640, 5150, 6150, and 8650 steels. (Table 2-9)
27. The blades of a post hole digger should have good wear resistance, high strength, and good ductility. SAE 1080 steel is a reasonable choice.
28. SAE 5160 OQT 1000 is a high-carbon, chromium steel, containing approximately 0.60% carbon and 0.80% chromium. It was heat treated by heating above its upper critical temperature, quenched in oil, and then tempered at 1000 degrees Fahrenheit. It has fairly high strength ($s_y = 151$ ksi or 1040 MPa) and good ductility (14% elongation).

29. In general, a high hardness with good ductility are desirable for machine parts and tools subjected to impact loads as seen by a shovel. A hardness of HRC 40 corresponds to approximately HB 375 and is considered moderately hard. While this is a good level, even a higher value up to HRC 50 (HB 475) would be better, provided ductility is fairly high, say about 15% elongation. Appendix 3 shows some forms of oil-quenched SAE 1040 and none listed have sufficiently high hardness. Appendix 4-1 shows the same material quenched in water and tempered. SAE 1040 WQT 700 has a hardness of HB 401 (HRC 43) with approximately 20% elongation and a yield point of 92 ksi.
30. Through hardening involves heating the entire part followed by quenching to achieve the hardened condition. Except for some variation in thick sections, the part is hardened throughout. But no chemical composition changes occur. In carburizing, the chemical composition of the surface is changed by the infusion of carbon. Thus, carburizing results in a hard surface while the core is softer.
31. Induction hardening is a heat treating process in which the area to be hardened is subjected to a high-frequency electric current created by a coil, inducing current flow near the surface of the part and causing local heating. After sufficient time to bring the surface to a temperature above the upper critical temperature of the material, the part is quenched to harden the surface.
32. Some carburizing grades of steels are SAE 1015, 1020, 1022, 1117, 1118, 4118, 4320, 4620, 4820, 8620 and 9310. The carbon content ranges from 0.10% to 0.20%. App. A-5.
33. The AISI 200 and 300 series of stainless steels are nonmagnetic.
34. Chromium gives stainless steels good corrosion resistance.
35. ASTM A992 structural steel is used for most wide-flange beams.
36. HSLA structural steels are high-strength, low-alloy steels having yield strengths in the range of 42 - 100 ksi (290 - 700 MPa).
37. Three types of cast iron are gray iron, ductile iron, and malleable iron.
38. ASTM A48 , Grade 30 is a gray iron with a tensile strength of 30 ksi (207 MPa); no yield strength; less than 1% elongation (brittle); modulus of elasticity (stiffness) of 16.9×10^6 psi (117 GPa).

Problem 38. (continued)

ASTM A536. Grade 100-70-03 is a ductile iron with a tensile strength of 100 ksi (689 MPa); a yield strength of 70 ksi (483 MPa); 3% elongation (brittle); modulus of elasticity (stiffness) of 24×10^6 psi (165 GPa).

ASTM A47, Grade 325/0 is a malleable iron with a tensile strength of 50 ksi (345 MPa); a yield strength of 32.5 ksi (224 MPa); 10% elongation (ductile); modulus of elasticity (stiffness) of 25×10^6 psi (172 GPa).

ASTM A220, Grade 70003 is a malleable iron with a tensile strength of 85 ksi (586 MPa); a yield strength of 70 ksi (483 MPa); 3% elongation (brittle); modulus of elasticity (stiffness) of 26×10^6 psi (179 GPa).

39. Powdered metals are preformed in a die under high pressure and sintered at a high temperature to fuse the particles. Re-pressing after sintering is sometimes used.
40. Parts made from Zamak 3 zinc casting alloy typically have good dimensional accuracy and smooth surfaces, a tensile strength of approximately 41 Ksi (283 MPa), a yield strength of 32 Ksi (221 MPa), 10% elongation, and a modulus of elasticity of 12.4×10^6 psi (85 GPa). (Appendix 10)
41. Type D tool steels are typically used for stamping dies, punches, and gages. (Table 2-11)
42. The suffix O on aluminum 6061-O indicates the annealed condition.
43. The suffix H on aluminum 3003-H14 indicates that it was strain hardened.
44. The suffix T on aluminum 6061-T6 indicates that it was heat treated.
45. Aluminum 7075-T6 has the highest strength; tensile strength = 80 ksi (60 MPa); yield strength = 70 ksi (53 MPa).
46. Aluminum alloy 6061 is one of the most versatile.
47. Three typical uses of titanium alloys are aerospace structures, chemical processing equipment, and marine hardware.
48. Bronze is an alloy of copper with tin, aluminum, lead, phosphorus, nickel, zinc, manganese, or silicon.

49. Bronze C86200 is a manganese bronze casting alloy with a tensile strength of 95 ksi (655MPa); yield strength of 48 ksi (331 MPa); 20% elongation (ductile); modulus of elasticity of 15×10^6 psi (103 GPa).
50. Bronze is used for gears and bearings.
51. Thermosetting plastics undergo a chemical change during forming resulting in a structure of cross-linked molecules. The process cannot be reversed or repeated. Thermoplastic materials can be formed repeatedly by reheating because the molecular structure is essentially unchanged during processing.
52. a) Gears: Nylon, polycarbonate, acetal, ^{PET,} polyurethane elastomer, phenolic. b) Helmets: ABS and polycarbonates. c) Transparent shield: Acrylic. d) Structural housing: PET, ABS, polycarbonate, acrylic, PVC, phenolic, polyester/glass composite. e) Pipe: ABS, PVC. f) Wheels: Polyurethane elastomer. g) Switch parts: polyimide, phenolic, PET.
53. Designers of parts to be made from composite materials can control 1) base resin, 2) reinforcing fibers, 3) amount of fibers, 4) orientation of fibers, 5) number of layers, 6) overall thickness, 7) orientation of layers, 8) combinations of types of materials.
54. Composite materials are comprised of two or more different materials, typically a resin reinforced by fibers.
55. Resins used for composites include polyesters, epoxies, polyimides, PHENOLICS (ALL THERMOSETS), THERMO PLASTICS: PE, PA, PEEK, PPS, PVC.
56. Reinforcing fibers used for composites are glass, boron, aramid, and carbon/graphite.
57. Sporting equipment is made from glass/epoxy, boron/epoxy, and graphite/epoxy composites.
58. Aerospace structures are made from glass/epoxy, boron/epoxy, graphite/epoxy, and aramid/epoxy composites.
59. Sheet molding compound is typically a glass/polyester composite.
60. SMC's are used for auto and truck body panels and large housings.
61. Reinforcing fibers are produced as continuous filaments, chopped fibers, roving, fabric, yarn, and mats.

62. Wet processing of composites involves the layup of fabric reinforcing sheets on a form, saturation of the sheets with the resin, and curing under heat and pressure.
63. Preimpregnated composite materials are produced with the resin already on the fibers in a convenient form, called a prepreg. The prepreg is layered onto the form and cured.
64. SMC's are preimpregnated fabric sheets formed in a mold and cured simultaneously under heat and pressure.
65. Pultrusion is a process of coating the fiber reinforcement as it is pulled through a heated die to produce a continuous form such as tubing, structural shapes, rod, and hat sections used to stiffen aircraft structures.
66. In the filament winding process, continuous filaments are placed around a mandrel in a controlled pattern and then cured. The process is used for pipe, pressure vessels, rocket motor cases, containers and enclosures.
67. Specific strength is the ratio of the strength of a material to its specific weight.
68. Specific stiffness is the ratio of the modulus of elasticity of a material to its specific weight.
69. Many composites have significantly higher values of specific strength and specific stiffness than metals.

70 - 73 refer to Figure 2-23 and Table 2-17.

General conclusions from Questions 70 - 73: The specific strengths of the metals listed range from 0.194×10^6 to 1.00×10^6 in, approximately a factor of 5.0. The specific stiffnesses are very nearly equal for all metals listed, approximately 1.0×10^8 in. The specific strengths of the composites listed range 1.87 to 4.88×10^6 in, much higher than any of the metals. Glass/epoxy has a specific stiffness about $2/3$ that of the metals. The other composites listed range from 2.2 to 8.3 times as stiff as the metals.

See Section 2-17 for answers to Questions 74 to 100.

Supplementary Problems – Chapter 2

1. Poisson's ratio: a) Carbon steel – 0.29; c) Lead – 0.43; e) Concrete – 0.10 to 0.25
2. See Section 2-2, subsection: Flexural Strength and Modulus, and Figure 2-5.
3. Erosive, abrasive, adhesive, fretting, surface fatigue
4. From Table 2-6: 14 alloys listed, Examples: ASTM A36, SAE 1018 HR or CD, SAE 1045 HR or CD, SAE 8620 CD.
5. From Table 2-6: SAE 304 and SAE 316
6. From Table 2-6: Six alloys listed, Examples: 2024-T4, 3003-H14, 6061-T6, 6063-T6
7. From Section 2-3: ASTM International, AISI, SAE
8. From Section 2-3: Aluminum Association
9. From Table 2-7: a) DIN 42CrMo4 or W-1.7225; b) BS 708A42; c) EN 42CrMo4; d) GB ML42CrMo4; e) JIS SCM 440H
10. From Table 2-7: a) DIN C45 or W-1.0503; b) BS 060A47; c) EN C45; d) GB 699-45; e) JIS S45C
11. From Table 2-7: a) DIN X6Cr17 or W-1.4016; b) BS 430S17; c) EN X6Cr17; d) GB ML1Cr17; e) JIS SUS430
12. From Table 2-7: a) DIN AlZnMgCu1.5 or W-3.4365; b) BS L.95, L.96; c) EN AlZn6MgCu
13. Water, brine, mineral oil, water-soluble polyalkylene glycol (PAG)
14. From Section 2-6: Fine steel or cast iron shot is projected at high velocity on critical surfaces to produce residual compressive stress that tends to improve the fatigue strength.
15. From Table 2-10: ASTM A27/A27M; A915/A915M; A128/A128M; A148/A148M
16. From Table 2-10: ASTM A757; ASTM A351; ASTM A216; ASTM A389
17. Carbide austempered ductile iron – used for: railroad rolling stock, earthmoving equipment, agricultural machinery, crushers
18. From Section 2-10: White iron is made by rapidly chilling a casting of gray iron or ductile iron during solidification. ASTM Standard A532 describes the process. Used to improve wear resistance for ball mills, crushers, mixing equipment, and material handling devices.
19. From Section 2-11: Powders are pressed to their basic form and then heated to sinter the powder particles into a strong solid.
20. From Section 2-11: Powders are compressed by a flexible membrane in a hermetic chamber to produce a high density; may be done cold or at elevated temperatures.
21. From Section 2-11: Metal powders are fed into an injection molding machine to form a green part that is then sintered to complete the solidification and bonding processes.
22. From Section 2-11: Metal powders are first pressed and sintered, then forged in a closed-die press to achieve final form and properties.
23. From Table 2-12: Carbon steel F-0008-HT, $s_u = 85$ ksi (590 MPa);

- Low-alloy steel FL-4405-HT, $s_u = 160$ ksi (1100 MPa);
 Diffusion-alloyed steel FD-0205-HT, $s_u = 130$ ksi (900 MPa);
 Sinter-hardened steel FLC-4608-HT, $s_u = 100$ ksi (690 MPa)
24. From Table 2-12: a) Nickel silver – CNZ-1818; $s_u = 20$ ksi (118 MPa)
 b) Bronze – CTG-1001; (no strength listed; used for bearings)
 c) Copper – C-0000; No strength listed; used for electrical applications
 d) Aluminum - $s_u = 32$ ksi (221 MPa)
 25. From Section 2-11: Projected surface area less than 50 in² (32 000 mm²)
 26. From Section 2-12: Aluminum casting alloys: 202, 222, 319, 360, 413, 444, 512, 535, 712, 771, 850, 852. Others available.
 27. From Section 2-12: Aluminum 2014, 2024, 6061
 28. From Section 2-13: Zinc *alloy No. 3* or *Zamak 3*.
 29. From Section 2-13 and Appendix A10-1: Zinc ZA-8, $s_u = 54$ ksi (374 MPa)
 ZA-12, $s_u = 59$ ksi (404 MPa); ZA-27, $s_u = 61$ ksi (421 MPa)
 30. From Section 2-14: Nickel-based alloys have good corrosion resistance and retain good levels of strength at high temperatures.
 31. From Section 2-15: a) Bearing bronze C93200l; b) Phosphor bronze C54400;
 c) Muntz metal C37000; d) Manganese bronze C86200;
 e) Copper-nickel-zinc alloy C96200; f) Manganese bronze C67500
 32. From Section 2-15: H-numbers indicate the degree of hardening by strain hardening methods; a) H04 – Full hard; b) H02 – ½ hard; c) H01 – 1/8 hard; d) H08 – Spring hard
 33. From Section 2-15: TD temper indicates – solution heat treated and cold worked
 34. From Section 2-15 and Figure 2-18: As the percent cold reduction increases, tensile and yield strengths increase and ductility as indicated by percent elongation decreases.
 10% cold work: $s_u = 133$ ksi (917 MPa), $s_y = 121$ ksi (834 MPa), 17% elongation
 40% cold work: $s_u = 154$ ksi (1062 MPa), $s_y = 142$ ksi (979 MPa), 1% elongation
 35. From Section 2-18: Metals, polymers, ceramics, glasses, elastomers, hybrids
 36. From Section 2-18: Foams, sandwich structures, honeycomb structures
 37. From Fig. 2-31: d) Metals, b) ceramics, g) composites, c) polymers, a) wood,
 h) rubbers/elastomers, f) foams
 38. From Fig. 2-32: b) ceramics, d) metals, g) composites, a) wood, c) polymers, f) foams,
 h) elastomers
 39. From Figure 3-32: (lightest to heaviest) e) foams, a) wood, h) elastomers, g) composites,
 d and b) Metals and ceramics (about equal)

CHAPTER 3 STRESS AND DEFORMATION ANALYSIS

Direct Tension and Compression

1. $\sigma = F/A$; $A = \pi (18^2 - 12^2)/4 = 141.4 \text{ mm}^2$
 $\sigma = 4500 \text{ N} / 141.4 \text{ mm}^2 = 31.8 \text{ N/mm}^2 = 31.8 \text{ MPa}$
 $\delta = \frac{PL}{EA} = \frac{(4500 \text{ N})(750 \text{ mm})}{(207 \times 10^9 \text{ N/mm}^2)(141.4 \text{ mm}^2)} \times \frac{10^6 \text{ mm}^2}{\text{m}^2} = 0.12 \text{ mm}$

2. $\sigma = F/A = 3500 \text{ N} / (\pi (10)^2 / 4) \text{ mm}^2 = 44.6 \text{ MPa}$

3. $\sigma = F/A = 20 \times 10^3 \text{ N} / (0.30) \text{ mm}^2 = 66.7 \text{ MPa}$

4. $\sigma = F/A = 860 \text{ LB} / (0.40 \text{ in})^2 = 5375 \text{ psi}$

5. $\sigma = F/A = 1900 \text{ LB} / \pi (0.375 \text{ in})^2 / 4 = 17,200 \text{ psi}$

6. $\sigma = P/A$; $A = (12 \text{ mm})^2 = 144 \text{ mm}^2$; $\sigma = 5000 \text{ N} / 144 \text{ mm}^2 = 34.7 \text{ MPa}$
 $\delta = \frac{PL}{EA} = \frac{(5000 \text{ N})(1650 \text{ mm})}{(E \text{ N/mm}^2)(144 \text{ mm}^2)} = \frac{57292}{E} \text{ mm}$
(ALL)

a) AISI 1020; $E = 207 \text{ GPa} = 207 \times 10^3 \text{ N/mm}^2$; $\delta = 0.277 \text{ mm}$

b) AISI 8650; $E = 207 \text{ GPa}$; $\delta = 0.277 \text{ mm}$

c) DUCTILE IRON; $E = 165 \text{ GPa}$; $\delta = 0.347 \text{ mm}$

d) ALUMINUM 6061-T6; $E = 69 \text{ GPa}$; $\delta = 0.830 \text{ mm}$

e) TITANIUM Ti-6AL-4V; $E = 114 \text{ GPa}$; $\delta = 0.503 \text{ mm}$

f) PVC; $E = 2410 \text{ MPa} = 2070 \text{ N/mm}^2$; $\delta = 27.7 \text{ mm}$

g) PHENOLIC; $E = 7580 \text{ MPa}$; $\delta = 7.56 \text{ mm}$

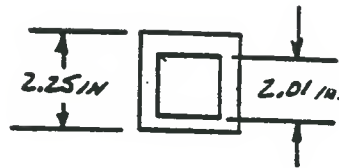
NOTE: STRESS IS CLOSE TO THE ULTIMATE FOR f AND g.

7.

$$\delta = PL/EA; P = \frac{\delta EA}{L}$$

$$A = (2.25^2 - 2.01^2) \text{ in}^2 = 1.02 \text{ in}^2$$

$$P = \frac{(0.004 \text{ in})(10 \times 10^6 \text{ lb/in}^2)(1.02 \text{ in}^2)}{16.0 \text{ in.}} = 2556 \text{ LB}$$



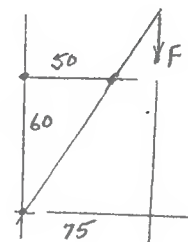
$$\sigma = P/A = 2556 \text{ LB} / 1.02 \text{ in}^2 = 2506 \text{ PSI}$$

8.

$$\sum M_B = 0 = 2500(75) - F_c(60)$$

$$F_c = 2500(75/60) = 3125 \text{ LB} = \text{TENSILE FORCE IN AC}$$

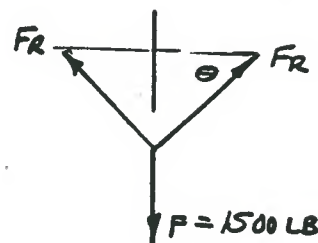
$$\sigma = \frac{P}{A} = \frac{3125 \text{ LB}}{(1.50)(3.50) \text{ in}^2} = 595 \text{ PSI}$$



9.

$$\sum F_R \sin \theta = 1500 \text{ LB}$$

$$F_R = 1500 / (2 \sin(45)) = 1061 \text{ LB}$$



10.

$$\sigma = P/A; \text{REQ'D. } A = P/\sigma_{\text{allow}}$$

$$A = \frac{1061 \text{ LB}}{18000 \text{ LB/in}^2} = 0.0589 \text{ in}^2 = \pi D^2/4$$

$$D = \sqrt{\frac{4A}{\pi}} = \sqrt{\frac{4(0.0589)}{\pi}} = 0.274 \text{ in. MINIMUM}$$

11.

$$\theta = 15^\circ; F_R = \frac{1500 \text{ LB}}{2 \sin \theta} = \frac{1500 \text{ LB}}{2 \sin(15)} = 2898 \text{ LB}$$

$$A = 2898 / 18000 = 0.161 \text{ in}^2; D = \sqrt{\frac{4(161)}{\pi}} = 0.453 \text{ in}$$

12.

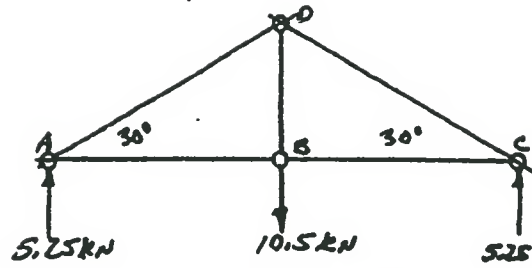
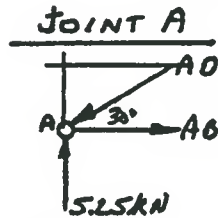
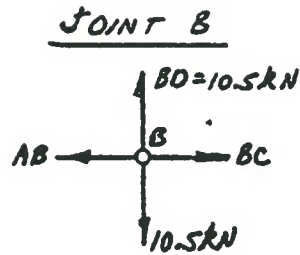


FIG. P8-12

$$\begin{aligned} AD \sin 30^\circ &= 5.25 \text{ kN} \\ AD &= 10.5 \text{ kN} = CD \\ AB &= AD \cos 30^\circ = 9.09 \text{ kN} = BC \end{aligned}$$

STRESSES:

$$AB, BC: \sigma_{AB} = \sigma_{BC} = \frac{9.09 \times 10^3 \text{ N}}{(12 \times 30) \text{ mm}^2} = \underline{25.3 \text{ MPa TENSION}}$$

$$BD: \sigma_{BD} = \frac{10.5 \times 10^3 \text{ N}}{(2 \times 10)(30) \text{ mm}^2} = \underline{17.5 \text{ MPa TENSION}}$$

$$AD, CD: A = (30)^2 - (20)^2 = 500 \text{ mm}^2$$

$$\sigma_{AD} = \sigma_{CD} = \frac{-10.5 \times 10^3 \text{ N}}{500 \text{ mm}^2} = \underline{-21.0 \text{ MPa COMPRESSION}}$$

13.

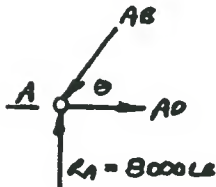
FIGURE P8-26

$$\sum M_A = 0 = 6000(6) + 12000(12) - R_F(18)$$

$$R_F = 10000 \text{ LB}$$

$$\sum M_F = 0 = 12000(6) + 6000(12) - R_A(18)$$

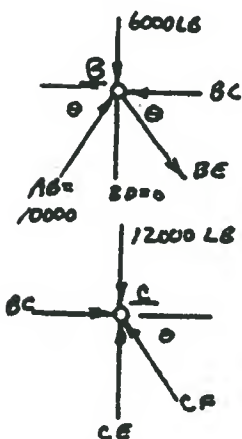
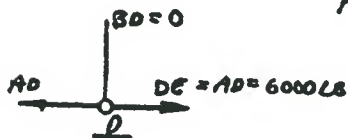
$$R_A = 8000 \text{ LB}$$



$$R_A = AB \sin \theta = AB(0.8)$$

$$AB = R_A / 0.8 = 8000 / 0.8 = 10000 \text{ LB COMP.}$$

$$AD = AB \cos \theta = 10000(0.6) = 6000 \text{ LB TENS.}$$



$$BE \sin \theta + 6000 - AB \sin \theta = 0$$

$$BE = \frac{AB \sin \theta - 6000}{\sin \theta} = \frac{10000(0.8) - 6000}{0.8} = 2500 \text{ LB TENS.}$$

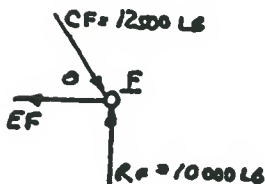
$$BC = AB \cos \theta + BE \cos \theta = 10000(0.6) + 2500(0.6) =$$

$$BC = 7500 \text{ LB COMP.}$$

$$BC = CF \cos \theta$$

$$CF = BC / \cos \theta = 7500 / 0.6 = 12500 \text{ LB COMP.}$$

$$CE = 12000 - CF \sin \theta = 12000 - 12500(0.8) = 2000 \text{ LB T}$$



$$EF = CF \cos \theta = 12500(0.6) = 7500 \text{ LB TENS.}$$

STRESSES:

$$\sigma_{AD} = \sigma_{DE} = 6000 / 0.968 = 6199 \text{ psi}$$

$$\sigma_{EF} = 7500 / 0.968 = 7748 \text{ psi}$$

$$\sigma_{BD} = 0$$

$$\sigma_{BE} = 2500 / 0.484 = 5165 \text{ psi}$$

$$\sigma_{CE} = 2000 / 0.484 = 4132 \text{ psi}$$

$$\sigma_{AB} = -10000 / 2.42 = -4132 \text{ psi}$$

$$\sigma_{BC} = -7500 / 2.42 = -3099 \text{ psi}$$

$$\sigma_{CF} = -12500 / 2.42 = -5165 \text{ psi}$$

AREAS OF MEMBERS: (APP. A5, A6)

$$AD, DE, EF - 2(0.484) = 0.968 \text{ IN}^2$$

$$BD, BE, CE - 0.484 \text{ IN}^2$$

$$AB, BC, CF - 2(1.21) = 2.42 \text{ IN}^2$$

NOTE: COMPRESSION MEMBERS MUST BE CHECKED FOR COLUMN BUCKLING

14.

$$A = (2.65)(1.40) + 2[(1.40)(0.5)(t)] = 4.41 \text{ IN}^2$$

$$\sigma = F/A = (52000 \text{ LB} / 4.41 \text{ IN}^2) = 11791 \text{ psi}$$

15.

$$A = (80)(40) + \pi(40)^2/4 = 4457 \text{ mm}^2$$

$$\sigma = F/A = 640 \times 10^3 \text{ N} / 4457 \text{ mm}^2 = \underline{143.6 \text{ MPa}}$$

Direct Shear Stress

16.

PIN DIA = 0.50 IN.; DOUBLE SHEAR

$$A_s = 2(\pi d^2/4) = 2 \pi (0.50)^2/4 = 0.3927 \text{ in}^2$$

$$\tau = F_s/A_s$$

MEMBER BC:

$$\sum M_B = 0 = (2500)(75) - F_{AC}(60)$$

$$F_{AC} = 2500(75/60) = 3125 \text{ LB}$$

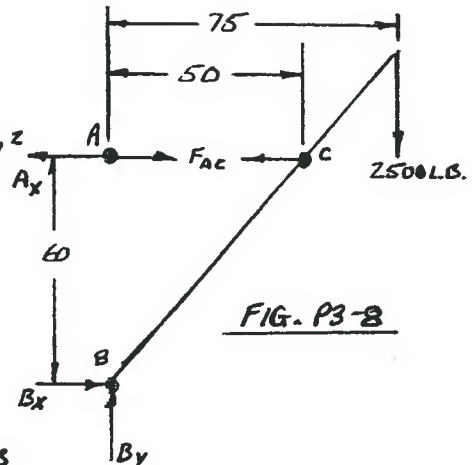
$$\text{AND } B_x = F_{AC} = 3125 \text{ LB}$$

$$B_y = 2500 \text{ LB}$$

$$\text{RESULTANT AT B: } B = \sqrt{3125^2 + 2500^2} = 4002 \text{ LB}$$

$$\text{PINS A AND C: } \tau = F_s/A_s = 3125 \text{ LB} / 0.3927 \text{ in}^2 = \underline{7958 \text{ PSI}}$$

$$\text{PIN B: } \tau = F_s/A_s = 4002 \text{ LB} / 0.3927 \text{ in}^2 = \underline{10,190 \text{ PSI}}$$



17.

FROM PROB. 9: FORCE IN EACH ROD = $F_R = 1500 \text{ LB} / 2 \sin \theta$

FOR $\theta = 40^\circ$; $F_R = 1167 \text{ LB} = \text{SHEAR FORCE ON UPPER PINS}$

ASSUME DOUBLE SHEAR: $A_s = 2 \pi d^2/4 = 2 \pi (0.75)^2/4 = 0.8836 \text{ in}^2$

$$\tau = F_s/A_s = 1167 \text{ LB} / 0.8836 \text{ in}^2 = \underline{1321 \text{ PSI}}$$

LOWER PIN: $F_s = 1500 \text{ LB}$

$$\tau = F_s/A_s = 1500 \text{ LB} / 0.8836 \text{ in}^2 = \underline{1698 \text{ PSI}}$$

18.

ANALYSIS FROM PROBLEMS 9 AND 17. LET $\theta = 15^\circ$

$$F_R = 1500 \text{ LB} / 2 \sin \theta = 1500 \text{ LB} / 2 \sin 15^\circ = 2898 \text{ LB}$$

$$\tau = F_s/A_s = 2898 \text{ LB} / 0.8836 \text{ in}^2 = \underline{3280 \text{ PSI}} \text{ IN ALL PINS}$$

19.

FIGURE 3-7 KEY IN SHEAR. $A_s = b \cdot L = (12)(45) = 540 \text{ mm}^2$

$$F_s = \text{TORQUE} / \text{RADIUS} = 1600 \text{ N} \cdot \text{m} / 30 \text{ mm} \times \frac{10 \text{ mm}}{\text{m}} = 53333 \text{ N}$$

$$\tau = F_s/A_s = \frac{53333 \text{ N}}{540 \text{ mm}^2} = 98.8 \text{ N/mm}^2 = \underline{98.8 \text{ MPa}}$$

20. PUNCH - FIG P3-20 $A_s = (\text{PERIM.}) t = [2.50 + 2.00 + 1.50 + \sqrt{0.5^2 + 2.5^2}] (0.060)$
 $A_s = (8.55 \text{ IN}) (0.060 \text{ IN}) = 0.513 \text{ IN}^2$
 $\tau = F_s / A_s = 52000 \text{ LB} / 0.513 \text{ IN}^2 = 101,400 \text{ PSI}$

21. PUNCH - FIG. P3-21. $\text{PERIM} = 60 + 2(30) + 2(7.5) + 3 \left[\frac{\pi(15)}{2} \right] = 205.7 \text{ mm}$
 $A_s = (\text{PERIM.}) t = (205.7)(2.0) = 411.4 \text{ mm}^2$
 $\tau = F_s / A_s = \frac{225000 \text{ N}}{411.4 \text{ mm}^2} = 547 \text{ N/mm}^2 = 547 \text{ MPa}$

Torsion

22. $\tau = \frac{T}{Z_p} = \frac{T}{\pi D^3 / 16} = \frac{800 \text{ N}\cdot\text{m}}{\pi (50)^3 / 16 \text{ mm}^3} \times \frac{10^3 \text{ mm}}{1 \text{ m}} = 32.6 \frac{\text{N}}{\text{mm}^2} = 32.6 \text{ MPa}$

23. $\theta = \frac{TL}{GJ} : T = 800 \text{ N}\cdot\text{m} = 800 \times 10^3 \text{ N}\cdot\text{mm} ; G = 80 \text{ GPa} = 80 \times 10^3 \text{ N/mm}^2$
 $J = \frac{\pi D^4}{32} = \frac{\pi (50)^4}{32} \text{ mm}^4 = 6.14 \times 10^5 \text{ mm}^4$
 $\theta = \frac{(800 \times 10^3 \text{ N}\cdot\text{mm})(850 \text{ mm})}{(80 \times 10^3 \text{ N/mm}^2)(6.14 \times 10^5 \text{ mm}^4)} = 0.0138 \text{ RAD} \times \frac{180 \text{ DEG}}{\pi \text{ RAD}} = 0.79 \text{ DEG.}$

24. $\tau = \frac{T}{Z_p} = \frac{T}{\pi D^3 / 16} = \frac{88.0 \text{ LB}\cdot\text{IN.}}{\pi (0.40 \text{ IN})^3 / 16} = 7003 \text{ PSI}$

25. $T = 63000 \text{ (P) / m} = 63000 (110 \text{ HP}) / 560 \text{ RPM} = 12375 \text{ LB}\cdot\text{IN.}$
 $\tau = \frac{T}{Z_p} = \frac{12375 \text{ LB}\cdot\text{IN.}}{\pi (1.25 \text{ IN})^3 / 16} = 32270 \text{ PSI}$

26. $T = P / \omega = 28 \times 10^3 \text{ N}\cdot\text{m} / \text{s} / 45 \text{ RAD/s} = 622 \text{ N}\cdot\text{m}$
 $Z_p = \frac{\pi (D^4 - d^4)}{16 D} = \frac{\pi [40^4 - 30^4] \text{ mm}^4}{16 (40 \text{ mm})} = 8590 \text{ mm}^3$
 $\tau = \frac{T}{Z_p} = \frac{622 \text{ N}\cdot\text{m}}{8590 \text{ mm}^3} \times \frac{10^3 \text{ mm}}{\text{m}} = 72.4 \text{ N/mm}^2 = 72.4 \text{ MPa}$

27. $\theta = \frac{TL}{GJ} : J = \frac{\pi (D^4 - d^4)}{32} = \frac{\pi [40^4 - 30^4]}{32} \text{ mm}^4 = 1.718 \times 10^5 \text{ mm}^4$
 $\theta = \frac{(622 \text{ N}\cdot\text{m})(400 \text{ mm})}{(80 \times 10^3 \text{ N/mm}^2)(1.718 \times 10^5 \text{ mm}^4)} \times \frac{10^3 \text{ mm}}{\text{m}} = 0.018 \text{ RAD} \times \frac{180^\circ}{\pi \text{ RAD}} = 1.04^\circ$

Noncircular Members in Torsion

28. SQUARE: $a = 25 \text{ mm}$; $Q = 0.208 a^3 = 3250 \text{ mm}^3$ } FIG. 3-6
 $k = 0.141 a^4 = 5.51 \times 10^4 \text{ mm}^4$

$$\tau = T/Q = (230 \text{ N}\cdot\text{m} / 3250 \text{ mm}^3) \frac{10^3 \text{ mm}}{\text{m}} = 70.8 \text{ N/mm}^2 = 70.8 \text{ MPa}$$

$$\theta = \frac{TL}{GK} = \frac{(230 \times 10^3 \text{ N}\cdot\text{mm})(650 \text{ mm})}{(80 \times 10^3 \text{ N/mm}^2)(5.51 \times 10^4 \text{ mm}^4)} = 0.0339 \text{ RAD} \times \frac{180^\circ}{\pi \text{ RAD}} = 1.94^\circ$$

29. $h/r = 0.60/1.50 = 0.40$ (FIG. P3-29)

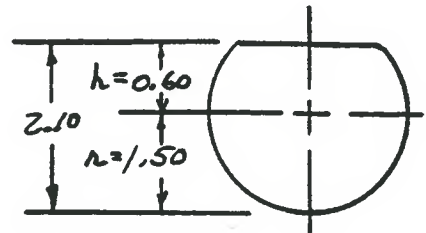
$$C_1 = 0.78; C_2 = 0.70$$

$$K = C_1 r^4 = 0.78 (1.50)^4 = 3.95 \text{ in}^4$$

$$Q = C_2 r^3 = 0.70 (1.50)^3 = 2.36 \text{ in}^3$$

$$\tau = \frac{T}{Q} = \frac{10600 \text{ LB}\cdot\text{in.}}{2.36 \text{ in}^3} = 4487 \text{ psi}$$

$$\theta = \frac{TL}{GK} = \frac{(10600 \text{ LB}\cdot\text{in.})(44.0 \text{ in})}{(11.5 \times 10^6 \text{ LB/in}^2)(3.95 \text{ in}^4)} = 0.0103 \text{ RAD} \times \frac{180^\circ}{\pi \text{ RAD}} = 0.59^\circ$$



30. $a = 2.0 \text{ in}$; $b = 4.0 \text{ in}$; $t = 0.109 \text{ in}$; $(a-t) = 1.891 \text{ in}$; $(b-t) = 3.891 \text{ in}$; $L = 6.5 \text{ FT}$
 $L = 78 \text{ in}$

$$K = \frac{2t(a-t)^2(b-t)^2}{a+b-2t} = \frac{2(0.109)(1.891)^2(3.891)^2}{[2.0+4.0-2(0.109)]} = 2.041 \text{ in}^4$$

$$Q = 2t(a-t)(b-t) = 2(0.109)(1.891)(3.891) = 1.604 \text{ in}^3$$

$$\tau = T/Q = (6000 \text{ LB/in}^2)(1.604 \text{ in}^3) = 9624 \text{ LB}\cdot\text{in}$$

$$\theta = \frac{TL}{GK} = \frac{(9624 \text{ LB}\cdot\text{in.})(78 \text{ in})}{(11.5 \times 10^6 \text{ LB/in}^2)(2.041 \text{ in}^4)} = 0.032 \text{ RAD} \times \frac{180^\circ}{\pi \text{ RAD}} = 1.83^\circ$$

Beams

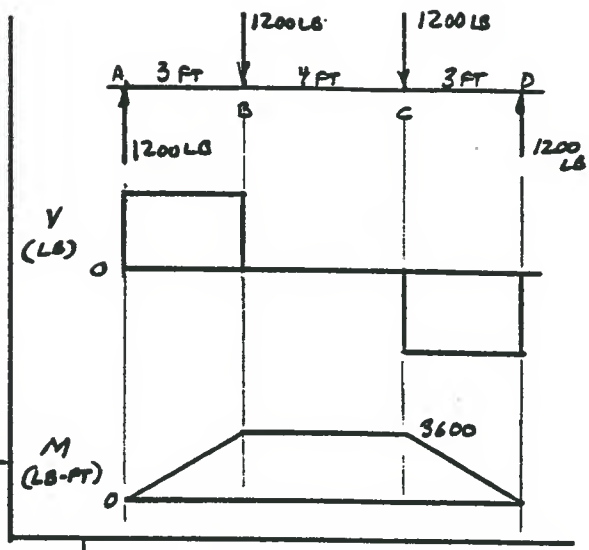
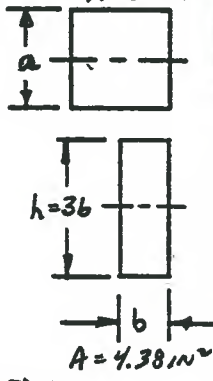
31.

$$\sigma = M/S : \text{REQ'D. } S = M/\sigma_{\text{ALLOW}}$$

$$S = \frac{3600 \text{ LB-FT}}{18000 \text{ LB/IN}^2} \times \frac{12 \text{ IN}}{\text{FT}} = 2.40 \text{ IN}^3 \quad A = 6.25 \text{ IN}^2$$

a) SQUARE; $S = a^3/6$
 $a = \sqrt[3]{6S} = 2.43 \text{ IN}$
 USE $a = 2.50 \text{ IN}$

b) RECTANGLE
 $S = \frac{bh^2}{6} = \frac{b(3b)^2}{6} = 1.5b^3$
 $b = \sqrt[3]{S/1.5} = 1.17 \text{ IN}$
 $h = 3b = 3.50 \text{ IN}$
 USE $b = 1.25 \text{ IN}, h = 3.50 \text{ IN}$

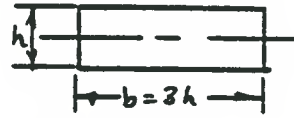


c) RECTANGLE

$$S = \frac{bh^2}{6} = \frac{3A(h)^2}{6} = \frac{A^2}{2}$$

$$h = \sqrt[3]{2S} = 1.69 \text{ IN}; b = 5.06 \text{ IN}$$

USE $h = 1.75 \text{ IN}; b = 5.00 \text{ IN} \quad A = 8.75 \text{ IN}^2$

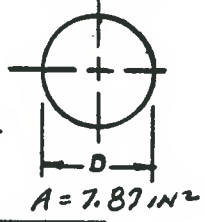


d) CIRCLE

$$S = \pi D^3/32$$

$$D = \sqrt[3]{\frac{32S}{\pi}} = 2.90 \text{ IN.}$$

USE $D = 3.00 \text{ IN.}$



e) $S_{4 \times 7.7}$
 $S = 3.04 \text{ IN}^3 \quad A = 2.26 \text{ IN}^2$

f) $C_{15 \times 40}$
 $S_y = 3.37 \text{ IN}^3 \quad A = 11.8 \text{ IN}^2$

g) 4-IN SCHEDULE 40 PIPE
 $S = 3.215 \text{ IN}^3 \quad A = 3.17 \text{ IN}^2$

32.

a) VOLUME = $A \times L = (2.5 \text{ IN})^2 \times 120 \text{ IN} = 750 \text{ IN}^3$; $W = (0.283 \text{ LB/IN}^3)(750 \text{ IN}^3)$

b) $V = A \times L = (1.25)(3.50)(120) = 525 \text{ IN}^3$; $W = (0.283)(525) = 149 \text{ LB}$

c) $V = (1.75)(5.00)(120) = 1050 \text{ IN}^3$; $W = (0.283)(1050) = 297 \text{ LB}$

d) $V = \frac{\pi D^2}{4} \times L = \frac{\pi (3.00 \text{ IN})^2}{4} \times 120 \text{ IN} = 848 \text{ IN}^3$; $W = (0.283)(848) = 240 \text{ LB}$

e) $7.7 \text{ LB/FT} (10 \text{ FT}) = 77.0 \text{ LB}$ (f) $40 \text{ LB/FT} (10 \text{ FT}) = 400 \text{ LB}$

g) $10.78 \text{ LB/FT} (10 \text{ FT}) = 107.8 \text{ LB}$
 $V = A \times L = (3.17 \text{ IN}^2)(120 \text{ IN}) = 38.09 \text{ IN}^3$
 FOR 1.0 FT
 $W = 0.283(38.09) = 10.78 \text{ LB/FT}$

33.

FROM CASE C; APPENDIX 14: $a = 36 \text{ IN}$; $L = 120 \text{ IN}$; $P = 1200 \text{ LB}$; $E = 30 \times 10^6 \text{ PSI}$

$$\eta_{\text{MAX}} = \frac{Pa}{24EI} (3L^2 - 4a^2) = \frac{1200(36)[3(120)^2 - 4(36)^2]}{24(30 \times 10^6)I} = \frac{2.281}{I} \text{ IN}$$

$$\eta_b = \eta_c = \frac{Pa^2(3L - 4a)}{6EI} = \frac{1200(36)^2[3(120) - 4(36)]}{6(30 \times 10^6)I} = \frac{1.866}{I} \text{ IN}$$

(SEE NEXT PAGE)

33.

(CONT)

$$a) I = a^4/12 = (2.50)^4/12 = 3.26 \text{ IN}^4 ; \gamma_{MAX} = \frac{2.281}{3.26} = 0.701 \text{ IN}$$

$$\gamma_B = \gamma_C = \frac{1.866}{3.26} = 0.572 \text{ IN.}$$

$$b) I = b h^3/12 = (1.25)(3.50)^3/12 = 4.47 \text{ IN}^4 ; \gamma_{MAX} = 0.511 \text{ IN} ; \gamma_B = 0.418 \text{ IN}$$

$$c) I = b h^3/12 = 5.00(1.75)^3/12 = 2.23 \text{ IN}^4 ; \gamma_{MAX} = 1.021 \text{ IN} ; \gamma_B = 0.836 \text{ IN.}$$

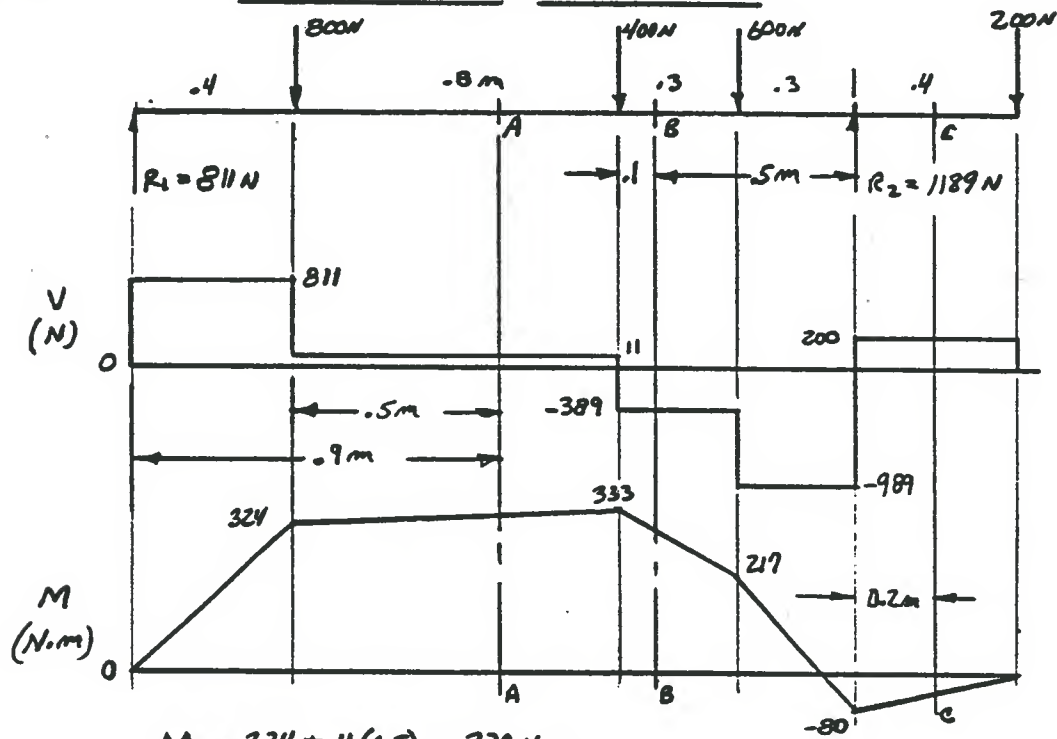
$$d) I = \frac{\pi D^4}{64} = \frac{\pi (3.00)^4}{64} = 3.98 \text{ IN}^4 ; \gamma_{MAX} = 0.574 \text{ IN} ; \gamma_B = 0.469 \text{ IN.}$$

$$e) I = 6.08 \text{ IN}^4 ; \gamma_{MAX} = 0.375 \text{ IN} ; \gamma_B = 0.307 \text{ IN.}$$

$$f) I_y = 9.23 \text{ IN}^4 ; \gamma_{MAX} = 0.247 \text{ IN} ; \gamma_B = 0.202 \text{ IN}$$

$$g) I = 7.23 \text{ IN}^4 ; \gamma_{MAX} = 0.315 \text{ IN} ; \gamma_B = 0.258 \text{ IN.}$$

34.



$$M_A = 324 + 11(0.5) = 330 \text{ N.m}$$

$$M_B = 333 - 389(0.1) = 294 \text{ N.m}$$

$$M_C = -80 + 200(0.2) = -40 \text{ N.m}$$

35.

FOR STRENGTH:

$$\text{REQ'D } S = \frac{M_{MAX}}{\sigma_{ALLOW}} = \frac{333 \text{ N.m}}{100 \text{ N/mm}^2} \times \frac{10^3 \text{ mm}}{\text{m}} = 3330 \text{ mm}^3 \text{ OR } 0.203 \text{ IN}^3$$

METRIC SHAPE WITH SMALLEST A AND $S \geq 3330 \text{ mm}^3$: APP. 16-19RES. Q: MECH. TUBING - ROUND: $D_o = 45 \text{ mm}$, $D_i = 40 \text{ mm}$, $A = 333.8 \text{ mm}^2$

$$S = 3361 \text{ mm}^3$$

36.

a) SIMPLE CANTILEVER - CASE 6 - APPENDIX 14-2 $I = 7.23 \text{ IN}^4$ APP. 15-17

$$M_A = \frac{-P X^2}{6EI} (3a - X) = \frac{-(800)(48)^2}{6(30 \times 10^6)(7.23)} [3(72) - 48] = -0.238 \text{ IN} = M_A$$

$$x_1 = 4 \text{ ft} = 48 \text{ IN}; a = 6 \text{ ft} = 72 \text{ IN}; x_2 = 8 \text{ ft} = 96 \text{ IN}.$$

$$M_B = \frac{-P a^2}{6EI} (3x_2 - a) = \frac{-800(72)^2}{6(30 \times 10^6)(7.23)} [3(96) - 72] = -0.688 \text{ IN} = M_B$$

b) SUPPORTED CANTILEVER - CASE 6 - APPENDIX 14-3

$$M_A = \frac{-P x_1^2 b}{12 E I L^3} (3 C_1 - C_2 x_1)$$

$$x_1 = 4 \text{ ft} = 48 \text{ IN}; a = 6 \text{ ft} = 72 \text{ IN}; b = 4 \text{ ft} = 48 \text{ IN}; L = 10 \text{ ft} = 120 \text{ IN}; N = 2 \text{ ft} = 24 \text{ IN}$$

$$C_1 = a L (L + b) = 72(120)(168) = 1.452 \times 10^6 \text{ IN}^3$$

$$C_2 = (L + a)(L + b) + a L = (92)(168) + 72(120) = 4.090 \times 10^4 \text{ IN}^2$$

$$M_A = \frac{-(800)(48)^2(48)}{12(30 \times 10^6)(7.23)(120)^3} [3(1.452 \times 10^6) - 4.090 \times 10^4(48)] = -0.047 \text{ IN} = M_A$$

$$M_B = \frac{-P a^2 N}{12 E I L^3} [3 L^2 b - N^2 (3L - a)]$$

$$M_B = \frac{-(800)(72)^2(24)}{12(30 \times 10^6)(7.23)(120)^3} [3(120)^2(48) - (24)^2(3(120) - 72)] = -0.042 \text{ IN} = M_B$$

37.

$$\sigma = M/S; S = M/\sigma_{\text{allow}}$$

$$S = \frac{8000 \text{ LB-IN}}{12000 \text{ LB/IN}^2} = 0.667 \text{ IN}^3$$

SMALLEST BEAM OK.

$$3 \text{ I} \times 1.637; S = 1.49 \text{ IN}^3; I = 2.24 \text{ IN}^4$$

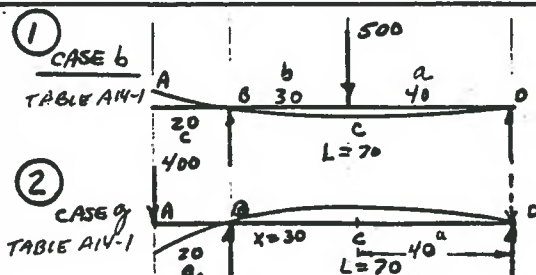
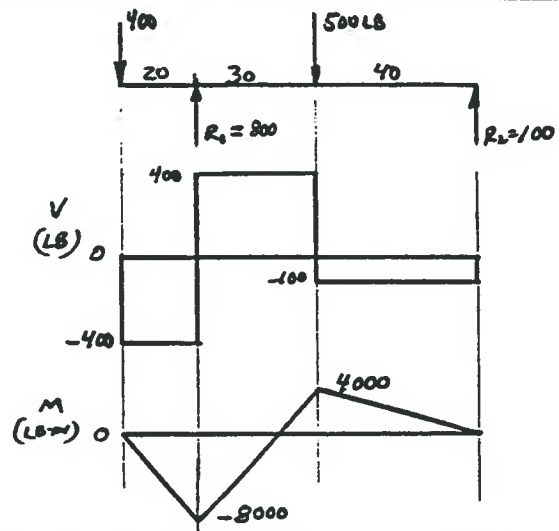
$$M_{A1} = \frac{+P a b c}{6 E I L} (L + a)$$

$$= \frac{+(500)(30)(40)(20)[70+40]}{6(10 \times 10^6)(2.24)(70)} = +0.140 \text{ IN UP}$$

$$M_{A2} = \frac{-P a^2}{3 E I} (a + L)$$

$$= \frac{400(20)^2(20+70)}{3(10 \times 10^6)(2.24)} = -0.214 \text{ IN DOWN}$$

$$M_A = M_{A1} + M_{A2} = +.140 - .214 = -0.074 \text{ IN DOWN}$$

SEE NEXT PAGE FOR M_C 

37. (CONT.) $\eta_{c1} = \frac{-Pa^2b^2}{3EI L} = \frac{-(500)(30)^2(40)^2}{3(10 \times 10^6)(2.24)(70)} = -0.153 \text{ IN DOWN}$

$\eta_{c2} = \frac{PaL^2}{EI} (0.06415) = \frac{(400)(20)(70)^2}{(10 \times 10^6)(2.24)} (0.06415) = 0.112 \text{ IN UP}$

NOTE: $a/L = 40/70 = 0.571$. THEN POINT C IS CLOSE TO η_{MAX} IN CASE g.

$\eta_c = \eta_{c1} + \eta_{c2} = -0.153 + 0.112 = -0.041 \text{ IN DOWN}$

38. $M_{\text{MAX}} = 3810 \text{ LB-FT} (121 \text{ N/FT})$

$= 45720 \text{ LB-IN}$

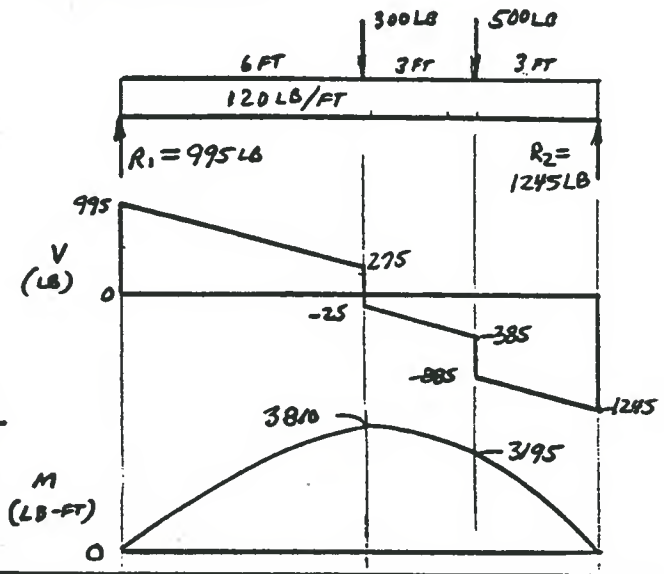
$I = \frac{bh^3}{12} = \frac{(150)(7.25)^3}{12} = 47.6 \text{ IN}^4$

$S = \frac{bh^2}{6} = \frac{(150)(7.25)^2}{6} = 13.1 \text{ IN}^3$

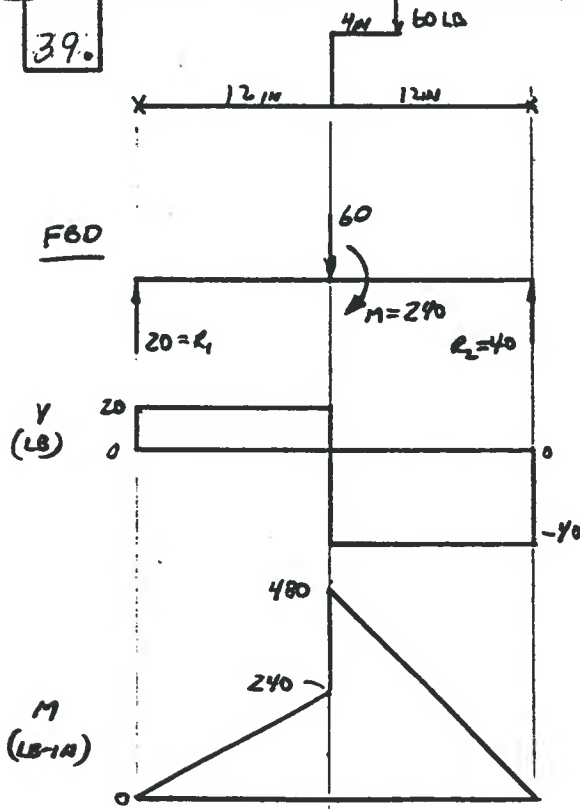
$\sigma = \frac{M}{S} = \frac{45720}{13.1} = 3480 \text{ PSI}$

$\tau = \frac{3V}{2A} = \frac{3(1245)}{2(150)(7.25)} = 172 \text{ PSI}$

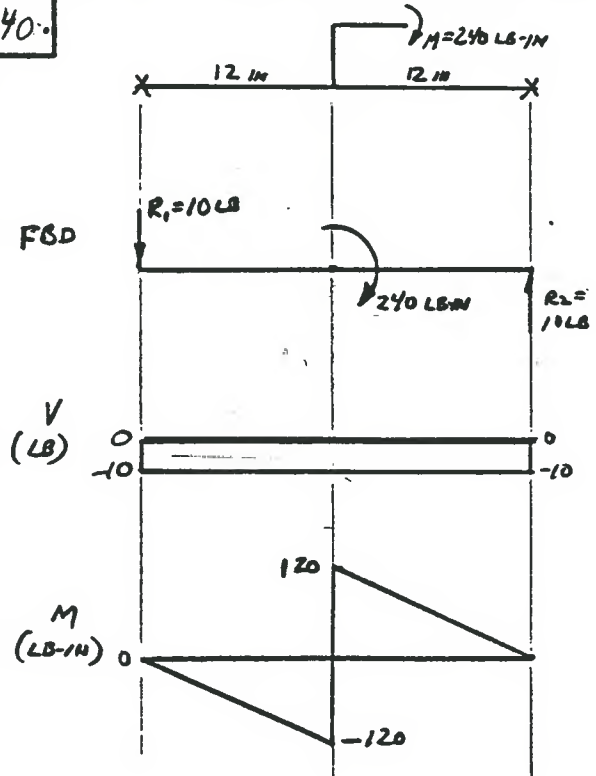
Beams with Concentrated Bending Moments



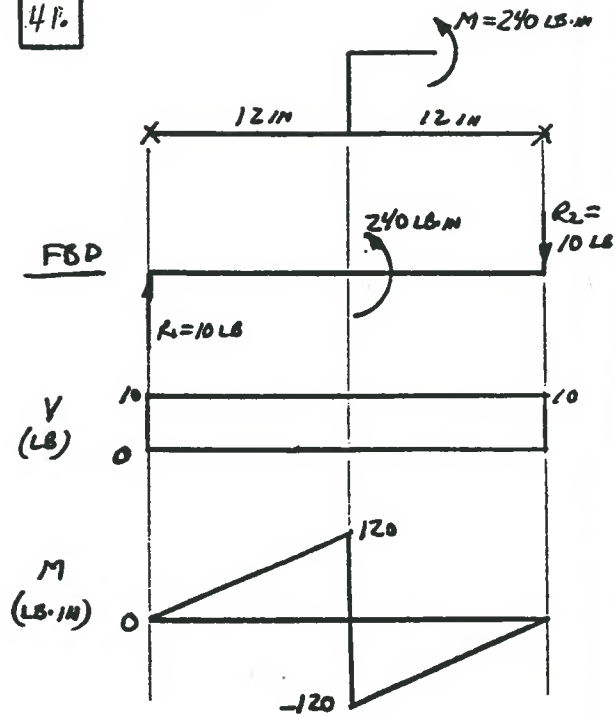
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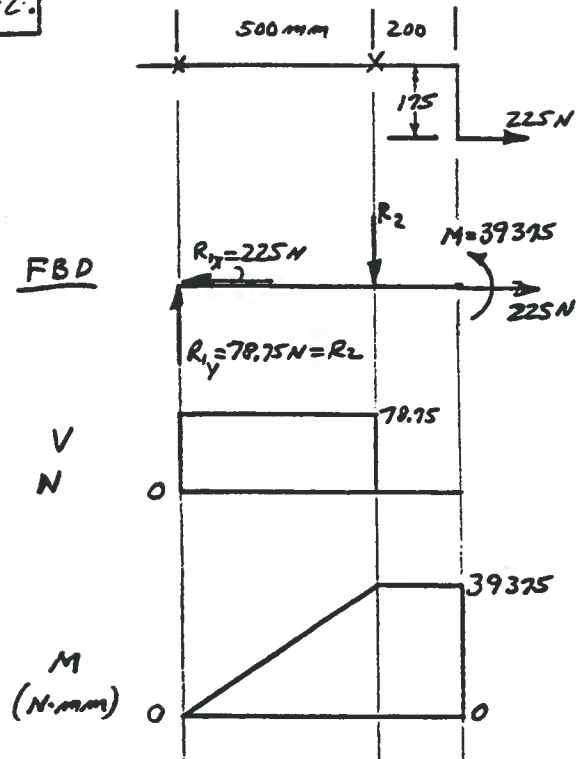
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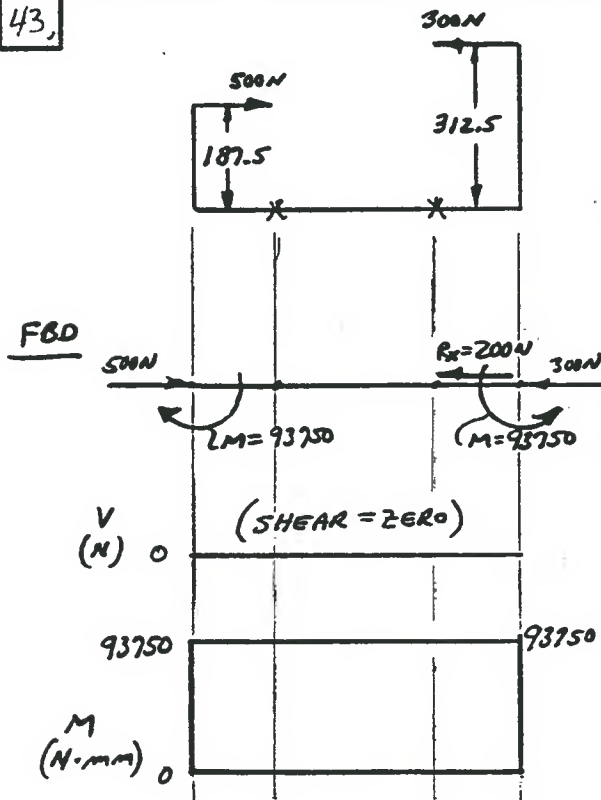
41.



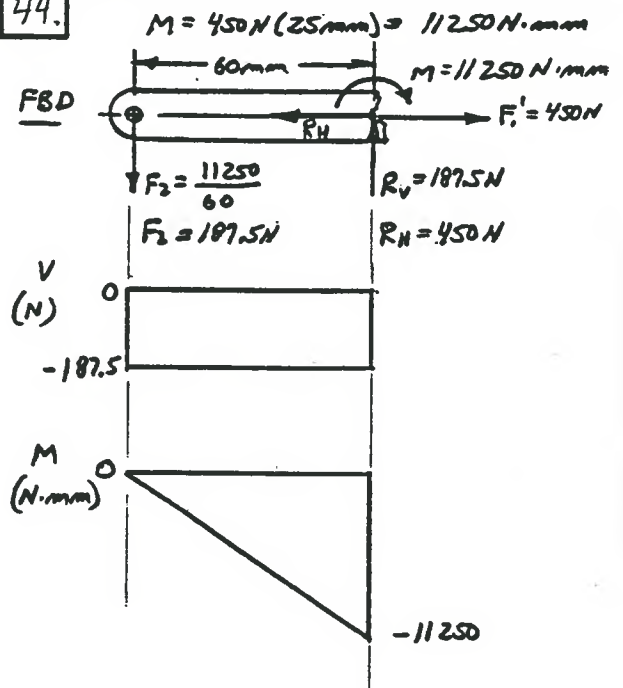
42.



43.

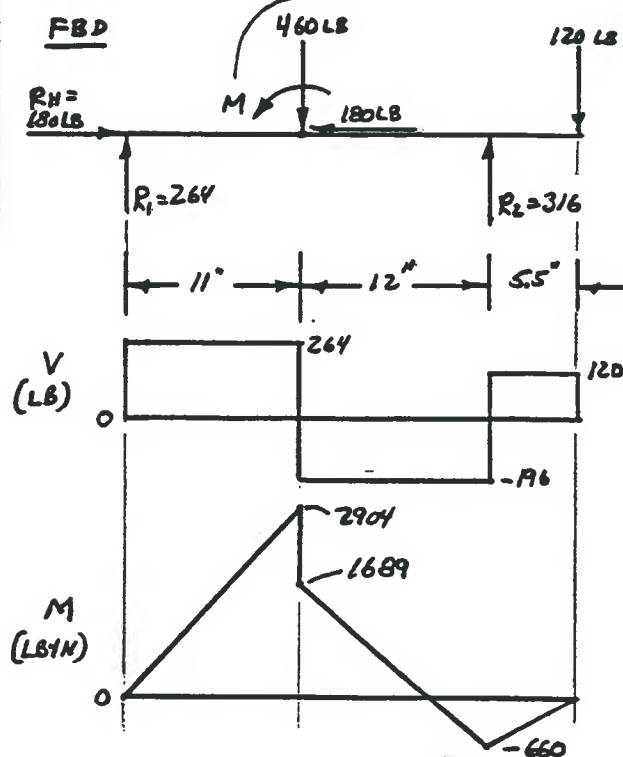


44.



45

$$M = 180(6.75) = 1215 \text{ LB}\cdot\text{IN}$$

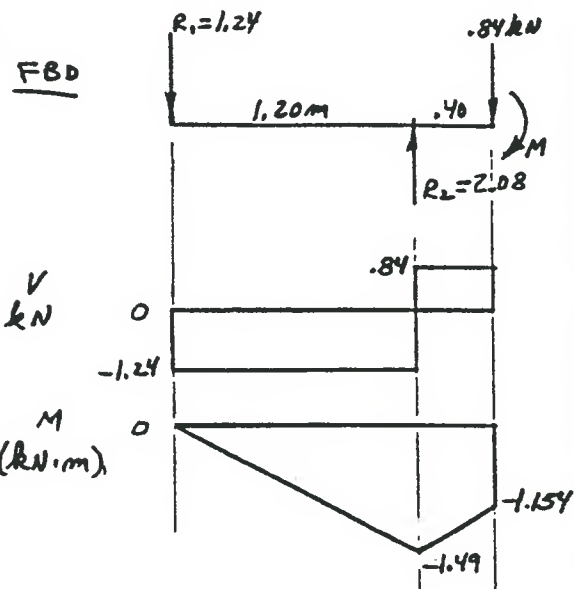


46

AT RIGHT END:

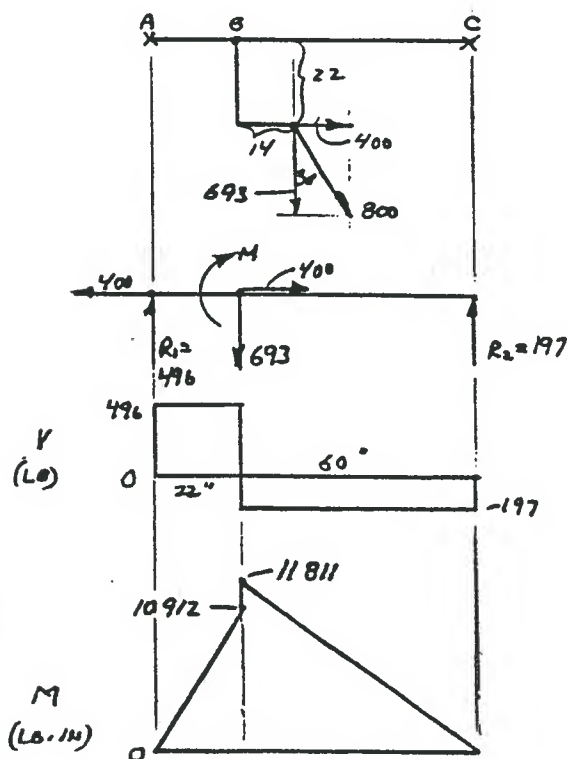
$$\text{NET VERTICAL FORCE} = 2.64 - 1.80 = 0.84 \text{ kN}$$

$$M = 2.64(2.60) + 1.80(.260) = 1.154 \text{ kN}\cdot\text{m}$$



47

$$M = 693(14) - 400(22) = 899 \text{ LB}\cdot\text{IN}$$

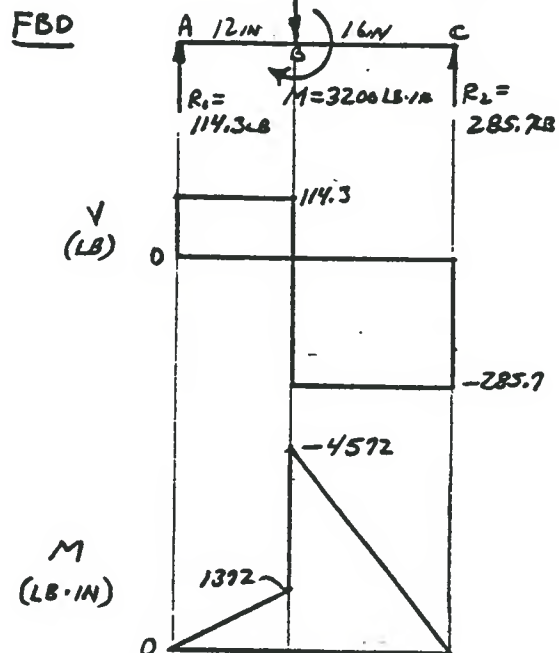


48

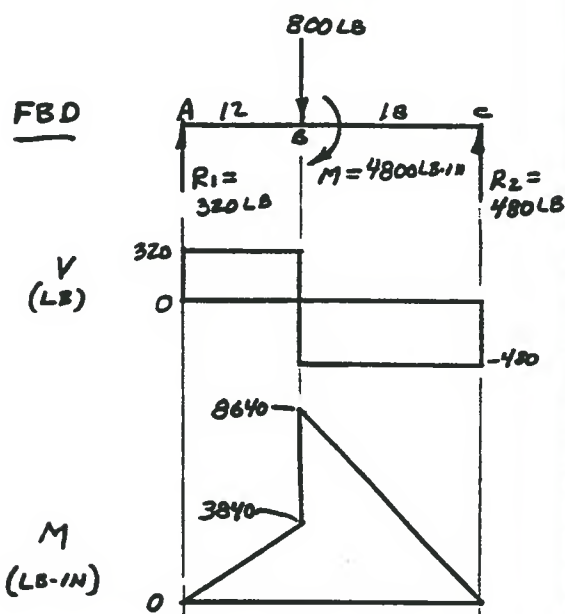
$$M = 200(4) + 600(4) = 3200 \text{ LB}\cdot\text{IN}$$

$$\text{TORQUE} = T = 200(8) + 600(8) = 6400 \text{ LB}\cdot\text{IN}$$

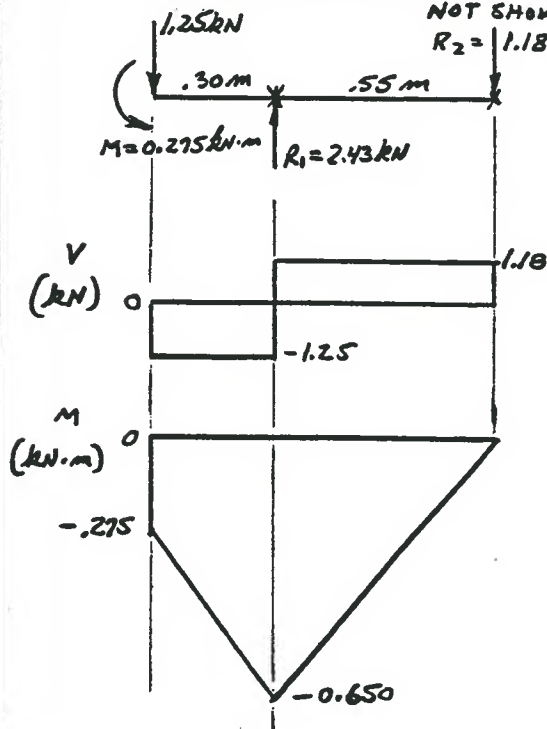
(NOT SHOWN)
CW - VIEWED FROM C.



49. $M = 600(6) + 200(6) = 4800 \text{ LB}\cdot\text{IN}$
 NET TORQUE = $600(4) - 200(10) = 400 \text{ LB}\cdot\text{IN}$
 NOT SHOWN

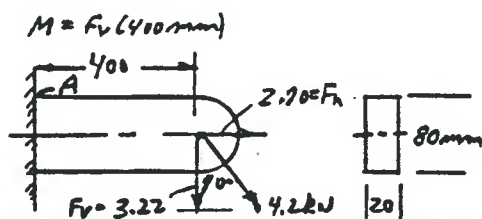


50. $M = 1.25 \text{ kN}(.22 \text{ m}) = 0.275 \text{ kN}\cdot\text{m}$
 $T = 1.25 \text{ kN}(.30 \text{ m}) = 0.375 \text{ kN}\cdot\text{m}$
 NOT SHOWN
 $R_2 = 1.18 \text{ kN}$

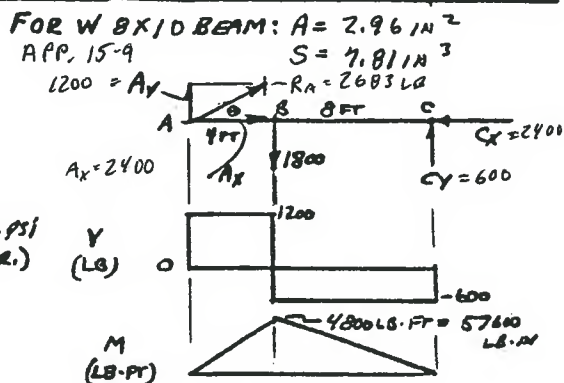


Combined Normal Stresses

51. $A = 20(80) = 1600 \text{ mm}^2$
 $S = bh^2/6 = 20(80)^2/6 = 21333 \text{ mm}^3$
 $\sigma_A = \frac{F_h}{A} + \frac{F_v(400)}{S} = \frac{2700 \text{ N}}{1600 \text{ mm}^2} + \frac{(3220 \text{ N})400 \text{ mm}}{21333 \text{ mm}^3}$
 $\sigma_A = 1.69 + 60.38 = 62.07 \text{ MPa}$



52. $\tan \theta = 4/8 = 0.5; \theta = 26.6^\circ$
 $\sum M_C = 0 = 1800(8) - A_y(12)$
 $A_y = 1200 \text{ LB}; C_y = 1800 - A_y = 600 \text{ LB}$
 $A_x = 2A_y = 2400 \text{ LB} = C_x$
 AT B ON TOP OF BEAM
 $\sigma = \frac{-A_x}{A} - \frac{M}{S} = \frac{-2400}{2.96} - \frac{57600}{7.81} = -8186 \text{ PSI}$
 AT B ON BOTTOM OF BEAM
 $\sigma = \frac{-A_x}{A} + \frac{M}{S} = \frac{-2400}{2.96} + \frac{57600}{7.81} = +6564 \text{ PSI}$
 (TENSILE)



53.

AT A (TOP)

$$\sigma = \frac{F_h}{A} + \frac{M}{S} = \frac{30 \text{ N}}{50(20) \text{ mm}^2} + \frac{(430 \text{ N})(400 \text{ mm})}{(20)(50)^2/6 \text{ mm}^3}$$

$$\sigma_A = 0.301 + 20.64 = 20.94 \text{ MPa TENSION}$$

AT B, $M = 430(200) = 86000 \text{ N}\cdot\text{mm}$ ASSUME AXIAL STRESS IS SMALL: $\sigma \approx M/S$

$$\text{REQ'D } S = \frac{M}{\sigma} = \frac{86000 \text{ N}\cdot\text{mm}}{20.94 \text{ N/mm}^2} = 4107 \text{ mm}^3 = th^2/6$$

$$h = \sqrt{\frac{6S}{t}} = \sqrt{\frac{6(4107)}{20}} = 35.1 \text{ mm} \quad \text{LET } h = 36 \text{ mm}; S = \frac{th^2}{6} = \frac{20(36)^2}{6} = 4320 \text{ mm}^3$$

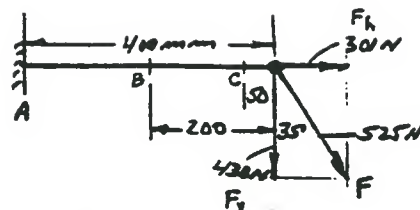
$$\sigma_B = \frac{30 \text{ N}}{720 \text{ mm}^2} + \frac{86000 \text{ N}\cdot\text{mm}}{4320 \text{ mm}^3} = 20.33 \text{ MPa OK}$$

AT C, $M = 430(50) = 21500 \text{ N}\cdot\text{mm}$

$$S = \frac{M}{\sigma} = \frac{21500}{20.94} = 1027 \text{ mm}^3; h = \sqrt{\frac{6S}{t}} = \sqrt{\frac{6(1027)}{20}} = 17.6 \text{ mm}$$

$$\text{LET } h = 18 \text{ mm}; t = 20 \text{ mm}; S = \frac{20(18)^2}{6} = 1080 \text{ mm}^3; A = 18(20) = 360 \text{ mm}^2$$

$$\sigma_C = \frac{30 \text{ N}}{360 \text{ mm}^2} + \frac{21500 \text{ N}\cdot\text{mm}}{1080 \text{ mm}^3} = 20.74 \text{ MPa OK}$$



54.

6x2 x 1/4 HOLLOW RECT. TUBE; APP. 15-14
FOR CROSS SECTION OF BEAM

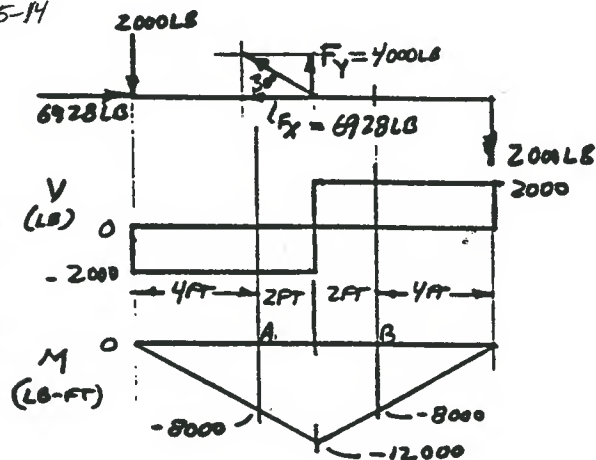
$$A = 3.59 \text{ in}^2$$

$$S = 4.60 \text{ in}^3$$

AT A: $\sigma = \frac{-F_x}{A} + \frac{M}{S}$ TENSION ON TOP SURFACE (LB)

$$\sigma_A = \frac{-6928 \text{ LB}}{3.59 \text{ in}^2} + \frac{8000(12) \text{ LB}\cdot\text{in}}{4.60 \text{ in}^3}$$

$$\sigma_A = -1930 \text{ PSI} + 20870 \text{ PSI} = 18940 \text{ PSI}$$

AT B: $\sigma = \frac{M}{S} = 20870 \text{ PSI}$ 

55.

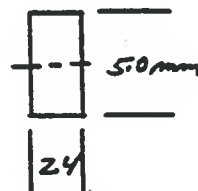
FROM FIG 3-22; $M = 875 \text{ N}\cdot\text{mm}$; $F_x = 35.0 \text{ N}$

$$A = 5.0(24) = 120 \text{ mm}^2$$

$$S = \frac{24(5.0)^2}{6} = 10.0 \text{ mm}^3$$

$$\sigma = \frac{-F_x}{A} + \frac{M}{S} = \frac{-35.0 \text{ N}}{12.0 \text{ mm}^2} + \frac{875 \text{ N}\cdot\text{mm}}{10.0 \text{ mm}^3} = 84.58 \text{ MPa TENSION}$$

ON BOTTOM OF SECTION



56.

$$BC = \sqrt{50^2 + 60^2} = 78.1 \text{ IN} \quad \left. \begin{array}{l} BF = 117.15 \\ CF = 25 \text{ IN} / \sin \theta = 39.05 \text{ IN} \end{array} \right\}$$

$$\Sigma M_B = 0 = -C_y(78.1) + F_y(117.15)$$

$$C_y = F_y(117.15/78.1) = 1728 \text{ LB}$$

$$C_x = C_y / \tan \theta = 1439 \text{ LB}$$

$$\Sigma M_C = 0 = 1152(39.05) - B_y(78.1)$$

$$B_y = 576 \text{ LB}$$

$$B_x = C_x + F_x = 1439 + 1383 = 2822 \text{ LB}$$

FOR 6x4 x 1/4 TUBE: (APP. 15-14)

$$A = 4.59 \text{ IN}^2; S_x = 7.36 \text{ IN}^3$$

JUST ABOVE C:

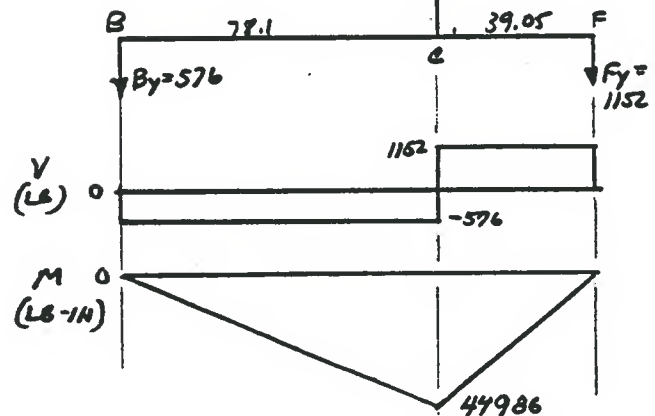
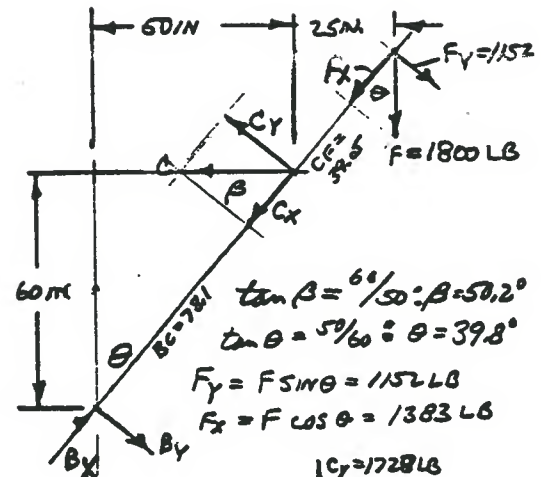
$$\sigma = \frac{-F_x}{A} + \frac{M}{S} = \frac{-1383}{4.59} + \frac{44986}{7.36}$$

$$\sigma = 5811 \text{ PSI TENSION}$$

JUST BELOW C:

$$\sigma = \frac{-B_x}{A} - \frac{M}{S} = \frac{-2822}{4.59} - \frac{44986}{7.36}$$

$$\sigma = -6727 \text{ PSI COMPRESSION}$$



57.

$$A = b^2; S = b^3/6$$

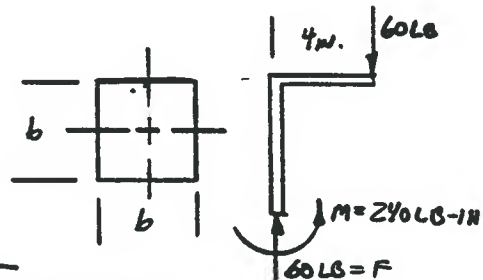
$$\sigma = -\frac{F}{A} - \frac{M}{S} = -\frac{60}{b^2} - \frac{240}{b^3/6}$$

$$\sigma = -\frac{60}{b^2} - \frac{1440}{b^3} \quad \text{BUT ASSUME } F/A \text{ IS SMALL}$$

$$\sigma \approx -1440/b^3 : b = \sqrt[3]{\frac{-1440}{\sigma_{\text{ALLOW}}}} = \sqrt[3]{\frac{-1440 \text{ LB-IN}}{-12000 \text{ LB/IN}^2}} = 0.493 \text{ IN}$$

$$\text{TRY } b = 0.500 \text{ IN} = 1/2 \text{ IN} : A = 0.25 \text{ IN}^2; S = b^3/6 = 0.0208 \text{ IN}^3$$

$$\sigma = \frac{-60 \text{ LB}}{0.25 \text{ IN}^2} - \frac{240 \text{ LB-IN}}{0.0208 \text{ IN}^3} = -240 - 11520 = -11760 \text{ PSI OK}$$



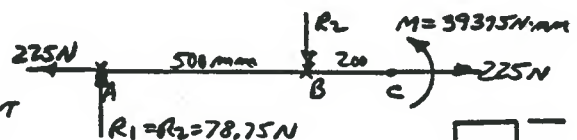
58.

FBD FROM PROBLEM 3-42:

PART FROM B-C SEES 39375 N-mm MOMENT

$$\sigma_{\text{max}} = \frac{F}{A} + \frac{M}{S} = \frac{225 \text{ N}}{2500 \text{ mm}^2} + \frac{39375 \text{ N-mm}}{20833 \text{ mm}^3}$$

$$\sigma_{\text{max}} = 0.09 + 1.89 = 1.98 \text{ MPa TENSION ON BOTTOM SURFACE BETWEEN B AND C.}$$



$$A = 50^2 = 2500 \text{ mm}^2$$

$$S = 50^3/6 = 20833 \text{ mm}^3$$



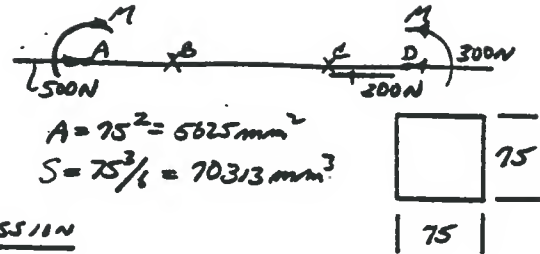
59. FROM PROB. 3-43: $M = \text{CONSTANT} = 93750 \text{ N}\cdot\text{mm}$
 $F_x = 500 \text{ N}$ FROM A TO C

$$\sigma_{\text{MAX}} = \frac{-F_x}{A} - \frac{M}{S}$$

$$= \frac{-500 \text{ N}}{5625 \text{ mm}^2} - \frac{93750 \text{ N}\cdot\text{mm}}{70313 \text{ mm}^3}$$

$$\sigma_{\text{MAX}} = -0.089 - 1.333 = \underline{7.422 \text{ MPa COMPRESSION}}$$

ON TOP SURFACE BETWEEN A AND C



60.

FROM PROB 3-47: POINT B IS WHERE BRACKET ATTACHES

TO RIGHT OF B; $M = 11811 \text{ LB}\cdot\text{IN}$; $\sigma = M/S$; $\text{REQ'D } S = \frac{M}{\sigma} = \frac{11811 \text{ LB}\cdot\text{IN}}{25000 \text{ LB/IN}^2} = 0.472 \text{ IN}^3$

$$S = \pi D^3/32$$

$$D = \sqrt[3]{32 \times 0.472 / \pi} = \sqrt[3]{32(0.472)/\pi} = 1.69 \text{ IN}$$

USE $D = 1.75 \text{ IN}$

CHECK TO LEFT OF B: $M = 10912 \text{ LB}\cdot\text{IN}$; $F_x = 400 \text{ LB}$

$$\sigma = \frac{F_x}{A} + \frac{M}{S} = \frac{400}{2.41} + \frac{10912}{0.526} = 166 + 20739 = 20905 \text{ PSI}$$

OK

Stress Concentrations: K_t factors obtained from the website:

www.efatigue.com/constantamplitude/stress-concentration/ [Note for a flat plate with a circular hole in bending: For $d/W < 0.50$, use $K_t = 1.0$. The eFatigue site does not show this.]

61. $D/d = 9/6 = 1.50$
 $r/d = 0.5/6 = 0.083$ } $K_t = 2.03$

$$\sigma_{\text{MAX}} = \frac{K_t F}{A} = \frac{(2.03)(1250 \text{ N})}{\frac{\pi(6)^2}{4} \text{ mm}^2} = \underline{89.7 \text{ MPa}}$$

62. $\sigma_{\text{MAX}} = K_t F/A$

BOTTOM: $D/d = 2.0/0.5 = 4.0$ } $K_t = 1.86$
 $r/d = 0.08/0.5 = 0.16$

$$\sigma_{\text{MAX}} = \frac{(1.86)(1200)}{\frac{\pi(0.5)^2}{4}} = 11,367 \text{ PSI MAX}$$

MIDDLE: $D/d = 2.0/0.75 = 2.67$ } $K_t = 2.08$
 $r/d = 0.08/0.75 = 0.11$

$$\sigma_{\text{MAX}} = \frac{(2.08)(2400)}{\frac{\pi(0.75)^2}{4}} = 11,300 \text{ PSI}$$

TOP: $D/d = 2.0/1.0 = 2.0$ } $K_t = 2.22$
 $r/d = 0.08/1.0 = 0.08$

$$\sigma_{\text{MAX}} = \frac{(2.22)(3600)}{\frac{\pi(1.0)^2}{4}} = 10,176 \text{ PSI}$$

63. LEFT HOLE: $d/W = 1.75/1.40 = 0.57$
 $K_t = 2.15$
 $\sigma_{\text{MAX}} = \frac{(2.15)(6200)}{(1.40 - 0.75)(0.5)} = 39,206 \text{ PSI MAX}$

MIDDLE: $d/W = 0.40/1.40 = 0.29$ } $K_t = 2.38$
 $\sigma_{\text{MAX}} = \frac{(2.38)(6200)}{(1.40 - 0.40)(0.5)} = 29,512 \text{ PSI}$

RIGHT: $d/W = 0.50/1.40 = 0.36$ } $K_t = 2.29$
 $\sigma_{\text{MAX}} = \frac{(2.29)(6200)}{(1.40 - 0.50)(0.5)} = 31,551 \text{ PSI}$

64. $H/h = 1.50/0.80 = 1.875$ } $K_t = 2.25$
 $r/h = 0.12/0.80 = 0.15$

$$\sigma_{\text{MAX}} = \frac{2.25(1625)}{(25)(0.80)} = 18,281 \text{ PSI}$$

65. $D/d = 42/30 = 1.40$ } $K_t = 2.30$
 $r/d = 1.50/30 = 0.05$

$$\sigma_{\text{MAX}} = \frac{(2.30)(30300 \text{ N})}{\frac{\pi(30)^2}{4} \text{ mm}^2} = 98.6 \text{ MPa}$$

66. $D/d = 2.00/1.25 = 1.60$ } $K_t = 1.43$
 $r/d = 0.10/1.25 = 0.08$

$$T_{\text{MAX}} = \frac{(1.43)(2200 \text{ LB}\cdot\text{IN})}{\frac{\pi(1.25)^3}{16} \text{ IN}^3} = 8203 \text{ PSI}$$

67.

$$\left. \begin{aligned} D/d &= 2.00/1.25 = 1.60 \\ h/b &= 0.06/1.25 = 0.048 \end{aligned} \right\} K_t = 2.23 \therefore \sigma_m = \frac{2.23 (2800) \text{ LB/IN}}{\pi (1.25)^3 / 32 \text{ IN}^3} = 32564 \text{ PSI}$$

68.

$$d/W = 1.38/2.00 = 0.69 \rightarrow K_t = 1.38$$

$$\sigma_m = K_t \sigma_{nom} = \frac{K_t 6MW}{(W^3 - d^3)t} = \frac{(1.38)(6)(12000)(2.00)}{[(2.00)^3 - (1.38)^3]0.75} = 49323 \text{ PSI}$$

Problems of a General Nature

69.

$$\sum M_C = 0 = 12.5 \text{ kN}(4.0 \text{ m}) - R_B(2.5 \text{ m})$$

$$R_B = (2.5)(4.0)/2.5 = 20.0 \text{ kN} \uparrow$$

$$R_C = 20 \text{ kN} - 12.5 \text{ kN} = 7.5 \text{ kN} \downarrow$$

$$M_{MAX} = 18.75 \text{ kN} \cdot \text{m} \times \frac{10^3 \text{ N}}{\text{kN}} \cdot \frac{10^3 \text{ mm}}{\text{m}}$$

$$M_{MAX} = 18.75 \times 10^6 \text{ N} \cdot \text{mm}$$

$$\text{SECTION MODULUS} = S = \frac{a^3}{6} = \frac{(20 \text{ mm})^3}{6}$$

$$S = 1333 \text{ mm}^3$$

$$\text{AREA OF AB} = (20 \text{ mm})^2 = 400 \text{ mm}^2$$

SHEAR AREA OF PIN - DOUBLE SHEAR

$$A_s = \frac{2\pi D^2}{4} = \frac{\pi D^2}{2} = \frac{\pi (8.0 \text{ mm})^2}{2} = 100.5 \text{ mm}^2$$

$$\text{TENSION IN AB: } \sigma = \frac{R_B}{A} = \frac{20000 \text{ N}}{400 \text{ mm}^2} = 50 \text{ MPa}$$

$$\text{SHEAR IN PIN: } \tau = \frac{R_B}{A_s} = \frac{20000 \text{ N}}{100.5 \text{ mm}^2} = 199 \text{ MPa}$$

$$\text{BENDING IN CD: AT B: } \sigma_B = M/S = \frac{18.75 \times 10^6 \text{ N} \cdot \text{mm}}{1333 \text{ mm}^3} = 14063 \text{ MPa}$$

VERY HIGH

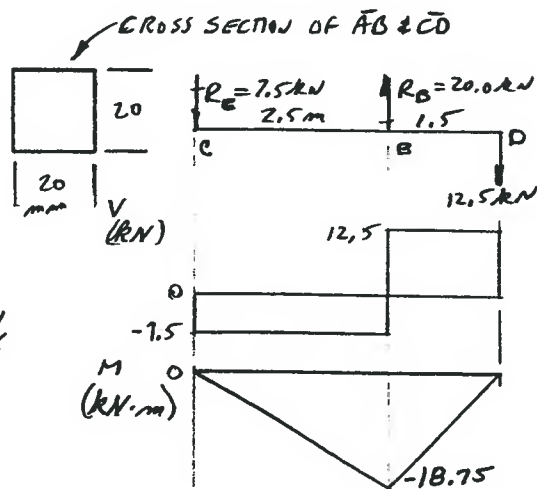
SUGGEST CHANGE IN CROSS SECTION OF CD TO MAKE $\sigma_B < 400 \text{ MPa}$

$$\text{REQ'D } S = M/\sigma = \frac{18.75 \times 10^6 \text{ N} \cdot \text{mm}}{400 \text{ N/mm}^2} = 46875 \text{ mm}^3$$

$$S = (20)(h)^2/6 : \text{REQ'D } h = \sqrt{6S/20} = \sqrt{\frac{6(46875 \text{ mm}^3)}{20 \text{ mm}}} = 118 \text{ mm}$$

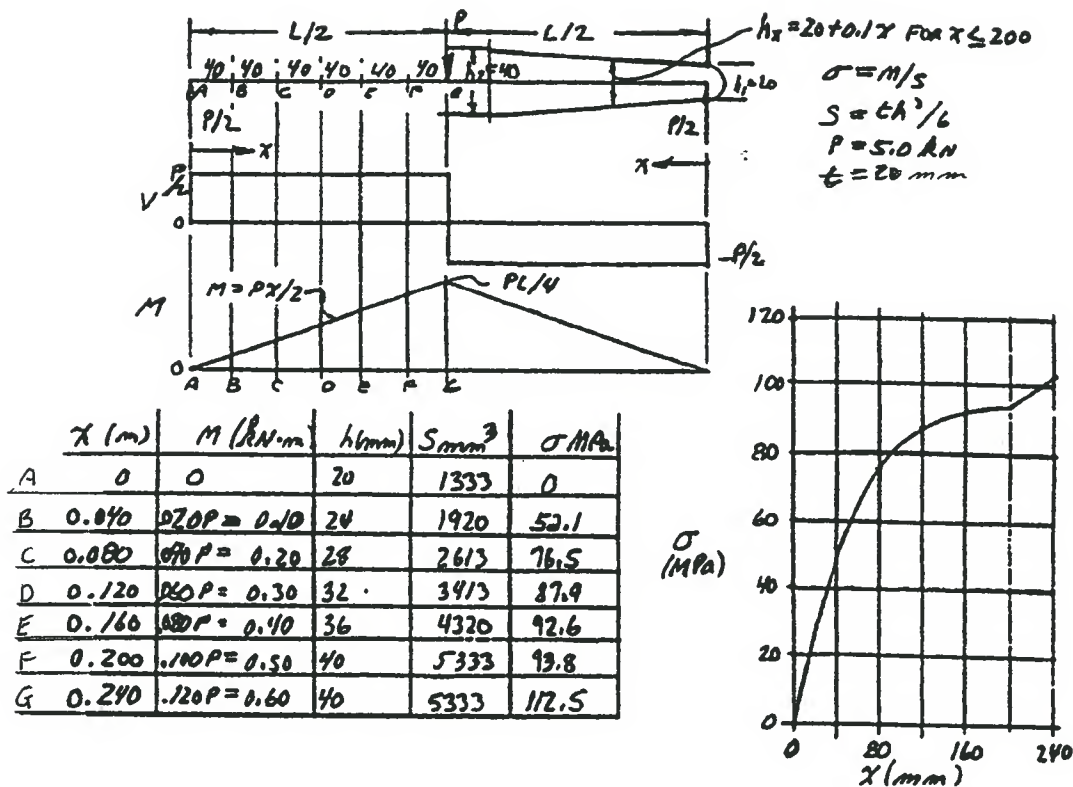
$$\text{LET } h = 120 \text{ mm: THEN } S = \frac{(20)(120)^2 \text{ mm}^3}{6} = 48000 \text{ mm}^3$$

$$\sigma = \frac{M}{S} = \frac{18.75 \times 10^3 \text{ N} \cdot \text{mm}}{48000 \text{ mm}^3} = 391 \text{ MPa OK}$$



PROPOSED CROSS SECTION OF CD

70.



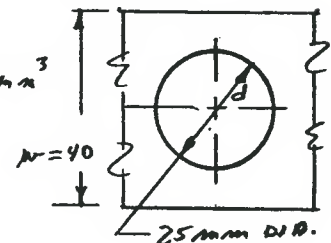
71.

FROM PROB. 70, $M = 0.60 \text{ kN} \cdot \text{m}$ UNDER LOAD.

$$\sigma = \frac{K_t M}{S_{NET}}; S_{NET} = \frac{(w^3 - d^3)t}{6w} = \frac{(40^3 - 25^3)(20)}{6(40)} = 4031 \text{ mm}^3$$

$$d/w = 25/40 = 0.625; K_t = 1.25$$

$$\sigma = \frac{(1.25)(0.60 \text{ kN} \cdot \text{m})}{4031 \text{ mm}^3} \cdot \frac{10^3 \text{ N}}{\text{kN}} \cdot \frac{10^3 \text{ mm}^3}{\text{m}^3} = 186 \text{ MPa}$$



72.

$$\sigma = \frac{K_t M}{S} \quad \text{ASSUME LATERAL BRACING}$$

(a) AT C AT LOAD; $M_C = 5143 \text{ LB} \cdot \text{FT}$

$$K_t = 1.0; S = \frac{th^3}{6} = \frac{(1.20)(4.0)^3}{6} = 3.20 \text{ in}^3$$

$$\sigma_C = \frac{(1.0)(5143 \text{ LB} \cdot \text{FT})(12 \text{ IN/FT})}{3.20 \text{ in}^3} = 19286 \text{ PSI}$$

(b) AT D AT 1.50 IN DIA. HOLE

$$M_D = 3857 \text{ LB} \cdot \text{FT}; d/w = \frac{1.50}{4.00} = 0.375$$

$$K_t = 1.00$$

$$S_{NET} = \frac{(w^3 - d^3)t}{6w} = \frac{(4.0^3 - 1.5^3)(1.20)}{6(4.0)} = 3.03 \text{ in}^3$$

$$\sigma_D = \frac{(1.0)(3857)(12)}{3.03} = 15270 \text{ PSI}$$

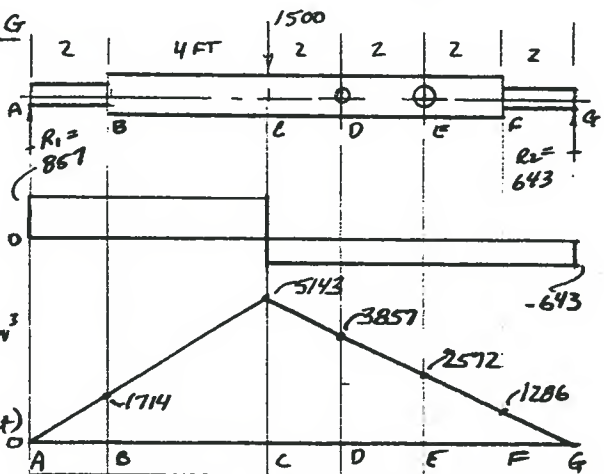
(c) AT E AT 2.50 IN DIA. HOLE

$$M_E = 2572 \text{ LB} \cdot \text{FT}; d/w = \frac{2.50}{4.00} = 0.625$$

$$K_t = 1.25$$

$$S_{NET} = \frac{(4.0^3 - 2.5^3)(1.20)}{6(4.00)} = 2.419 \text{ in}^3$$

$$\sigma_E = \frac{(1.25)(2572)(12)}{(2.419)} = 15948 \text{ PSI}$$

(d) AT B AT STEP; $H = 4.00$, $h = 2.80$, $r = 0.15$

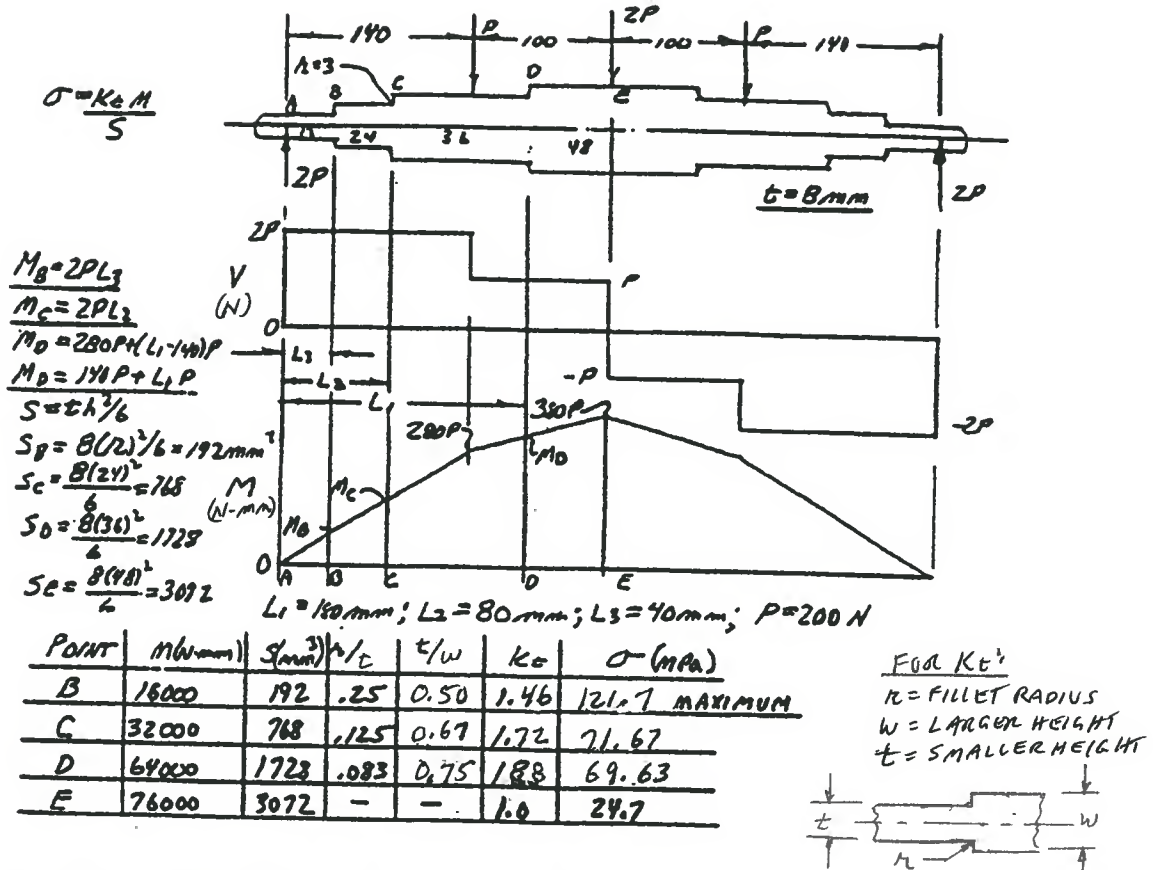
$$h/H = 1.43; r/H = 0.054; K_t = 2.21$$

$$M_B = 1714 \text{ LB} \cdot \text{FT}; S = \frac{th^3}{6} = \frac{(1.2)(2.8)^3}{6} = 1.57 \text{ in}^3$$

$$\sigma_B = \frac{(2.21)(1714)(12)}{(1.57)} = 29952 \text{ PSI}$$

MAXIMUM 1.57

73.



74.

$$M = F(S_2 + 2S_1/2) = (2500 \text{ N})(64.5 \text{ mm}) = 161250 \text{ N}\cdot\text{mm} \text{ ALONG UPPER PART}$$

$$\sigma = \frac{K_t M}{S_{NET}}; S_{NET} = \frac{(w^3 - d^3)(t)}{6w}$$

AT B-B: $d/w = 15/25 = 0.6 \rightarrow K_t = 1.20$

$$\sigma = \frac{(1.20)(6)(161250)(25)}{(25^3 - 15^3)(16)} = 148.1 \text{ MPa}$$

NOTE: For K_t at hole in flat plate in bending:
 If $d/W < 0.50$, use $K_t = 1.0$

75.

SEE ALSO PROBLEM 74.

AT B-B: $d/w = 12/25 = 0.48 \rightarrow K_t = 1.0$

$$\sigma = \frac{K_t M}{S_{NET}} = \frac{(1.0)(6)(161250)(25)}{(25^3 - 12^3)(16)} = 108.8 \text{ MPa}$$

76.

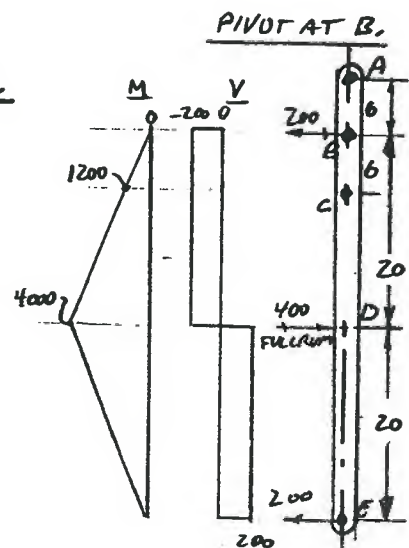
AT FULLRUM: $S = \frac{th^3}{6} = \frac{(0.75)(2.0)^3}{6} = 0.50 \text{ IN}^3$

$$\sigma_b = \frac{M}{S} = \frac{4000 \text{ LB}\cdot\text{IN}}{0.50 \text{ IN}^3} = 8000 \text{ PSI}$$

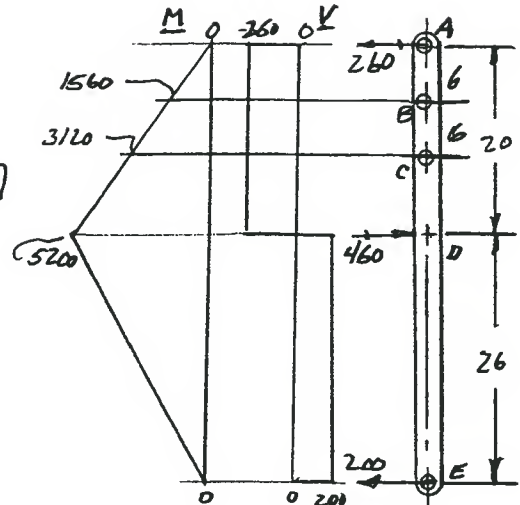
AT HOLE: APP 15-2: $d/w = 1.25/2.0 = 0.625$; $K_t = 1.25$

$$S_{NET} = \frac{(w^3 - d^3)t}{6w} = \frac{(2.00^3 - 1.25^3)(0.75)}{6(2.00)} = 0.378 \text{ IN}^3$$

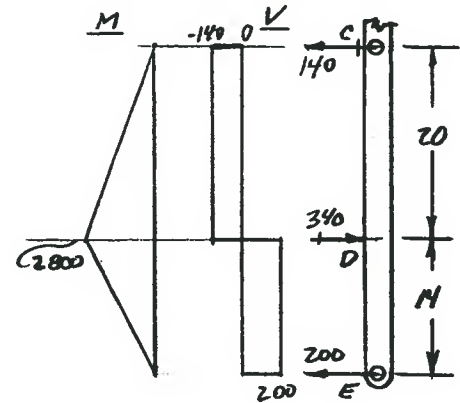
$$\sigma_c = \frac{K_t M_c}{S} = \frac{1.25(1200)}{0.378} = 3969 \text{ PSI}$$



71. PIVOT AT A.
AT FULCRUM: $S = \frac{bh^3}{6} = \frac{(0.75)(2.00)^3}{6} = 0.50 \text{ IN}^3$
 $\sigma_D = \frac{M}{S} = \frac{5200 \text{ LB}\cdot\text{IN}}{0.50 \text{ IN}^3} = 10,400 \text{ PSI}$
AT B: $K_t = 1.25$; $S_{\text{NET}} = 0.378 \text{ IN}^3$ [PR0076.]
 $\sigma_B = \frac{K_t M}{S} = \frac{1.25(1560)}{0.378} = 5160 \text{ PSI}$
AT C:
 $\sigma_C = \frac{K_t M}{S} = \frac{1.25(3120)}{0.378} = 10320 \text{ PSI}$

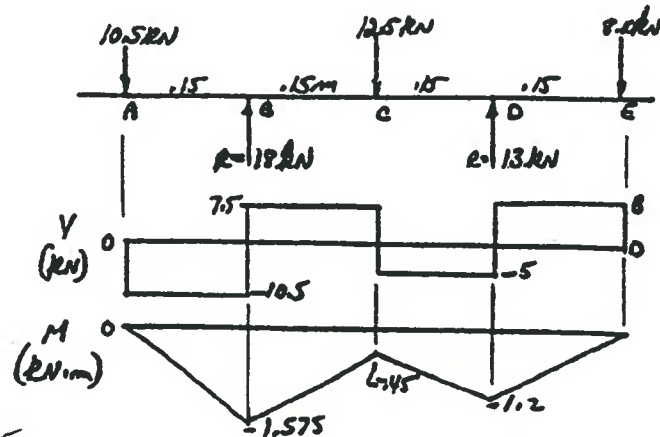


PIVOT AT E.
AT FULCRUM: $S = 0.50 \text{ IN}^3$
 $\sigma_D = \frac{M_D}{S} = \frac{2800 \text{ LB}\cdot\text{IN}}{0.50 \text{ IN}^3} = 5600 \text{ PSI}$



78.

$M = 0$ AT A, E
 POINT B IS CRITICAL
 $\sigma = \frac{MK_x}{S}$
 $S = \frac{\pi(YS)^3}{32} = 8946 \text{ mm}^3$
 FOR FILLET
 $\frac{r}{d} = \frac{2}{45} = 0.044$
 $\frac{D}{d} = \frac{55}{45} = 1.22$
 $K_x = 2.05$



$\sigma = \frac{MK_x}{S} = \frac{1.575 \times 10^3 \text{ N}\cdot\text{m} (2.05)}{8946 \text{ mm}^3} \times 10^3 \frac{\text{mm}^3}{\text{m}} = 360.9 \text{ MPa}$

79.

ONE POSSIBLE DESIGN

LUG JOINT: SEE FIG. 3-69; $d = 8.0 \text{ mm}$, $w = 20.0 \text{ mm}$
 FIND: THICKNESS t , MATERIALS FOR LUG AND PIN. $N = 5$.
 FROM SOLUTION FOR PROB. 3-69, $F = 20,000 \text{ N}$.

$$\text{LET } t = 0.5d = 0.5(8.0 \text{ mm}) = 4.0 \text{ mm}$$

$$\text{LET } w = d/0.40 = 8.0 \text{ mm}/0.40 = 20.0 \text{ mm}$$

$$\text{LET END DISTANCE} = h = w = 20.0 \text{ mm}$$

$$\text{HOLE DIAMETER} = d_{\text{hole}} = d_{\text{pin}}(1.002) = 8.016 \text{ mm}$$

$$\sigma_{\text{MAX}} = K_t \sigma_{\text{NOM}}; \sigma_{\text{NOM}} = \frac{F}{(w-d)t} = \frac{20000 \text{ N}}{(20-8)(4.0 \text{ mm})} = 416.7 \text{ MPa}$$

$$\text{IN FIG 3-29: } d/w = 8/20 = 0.40; K_t = 3.00$$

$$\sigma_{\text{MAX}} = K_t \sigma_{\text{NOM}} = (3.00)(416.7 \text{ MPa}) = 1250 \text{ MPa}$$

$$\text{LET } \sigma_{\text{MAX}} = \sigma_d = \frac{S_u}{N}; \text{REQ. } S_u = N \sigma_{\text{MAX}} = 5(1250) = 6250 \text{ MPa}$$

TOO HIGH FOR TYPICAL STEELS IN APP. 3.

RE-DESIGN FOR USE OF SAE 4340 OQT 800, $S_u = 1450 \text{ MPa}$

$$\sigma_d = \frac{S_u}{N} = \frac{1450 \text{ MPa}}{5} = 290 \text{ MPa}$$

$$\text{LET } \sigma_d = \sigma_{\text{MAX}} = K_t \sigma_{\text{NOM}} = 3.0 \sigma_{\text{NOM}} \text{ FOR } d/w = 0.40$$

$$\sigma_{\text{NOM}} = \frac{\sigma_d}{3.0} = \frac{290 \text{ MPa}}{3} = 96.67 \text{ MPa} = \frac{F}{A_{\text{NET}}} = \frac{F}{(w-d)t}$$

$$\text{REQ'D } A_{\text{NET}} = \frac{F}{96.67 \text{ MPa}} = \frac{20000 \text{ N}}{96.67 \text{ N/mm}^2} = 206.9 \text{ mm}^2$$

$$\text{FOR } w = \frac{d}{0.40} = 2.5d; t = 0.5d;$$

$$A_{\text{NET}} = (w-d)t = (2.5d-d)(0.5d) = 0.75d^2 = 206.9 \text{ mm}^2$$

$$d = \sqrt{206.9 \text{ mm}^2 / 0.75} = 16.6 \text{ mm.}$$

SPECIFY PREFERRED SIZE: $d = 18 \text{ mm}$ (APP. 2)

$$\text{THEN: } w = \frac{d}{0.4} = \frac{18}{0.4} = 45.0 \text{ mm}; t = 0.5d = 9.0 \text{ mm}$$

$$\sigma_{\text{NOM}} = \frac{F}{(w-d)t} = \frac{20000 \text{ N}}{(45-18)(9)} = \frac{20000 \text{ N}}{243 \text{ mm}^2} = 82.3 \text{ MPa}$$

(GIVEN DESIGN PARAMETERS WERE NOT FEASIBLE.)

$$\text{CHECK SHEAR STRESS IN PIN: } \tau = \frac{F}{A_s} = \frac{F}{2A} = \frac{F}{2(\pi d^2/4)} = \frac{F}{\pi d^2/2}$$

$$\tau = \frac{20000 \text{ N}}{\pi(18 \text{ mm})^2/2} = 39.3 \text{ MPa}$$

$$\text{CHECK } \tau_d = \frac{S_{su}}{N} = \frac{0.75 S_u}{5} = \frac{0.75(1450 \text{ MPa})}{5} = 217.5 \text{ MPa} > 39.3 \text{ MPa}$$

OK FOR PIN SHEAR

$$\text{HOLE DIA.} = d_{\text{HOLE}} = d_{\text{PIN}}(1.002) = 18.0 \text{ mm}(1.002) = 18.036 \text{ mm}$$

81

CURVED BEAMS: FIND F FOR YIELDING OF STEEL

ASTM A36 STEEL $S_y = 36 \text{ ksi} = 248 \text{ MPa}$

$$\sigma_o = \frac{M(R - r_o)}{A r_o (r_c - R)} \quad \sigma_i = \frac{M(R - r_i)}{A r_i (r_c - R)}$$

$$R = A / ASF; \quad A = b^2 = 10^2 = 100 \text{ mm}^2$$

$$\text{GIVEN: } r_i = 150 \text{ mm}; \quad r_o = r_i + 10 = 160 \text{ mm}$$

$$r_c = r_i + b/2 = 150 + 5 = 155 \text{ mm}$$

$$ASF = b \ln(r_o/r_i) = 10 \ln(160/150) = 0.64539 \text{ mm}$$

$$R = A / ASF = 100 / 0.64539 = 154.946 \text{ mm}$$

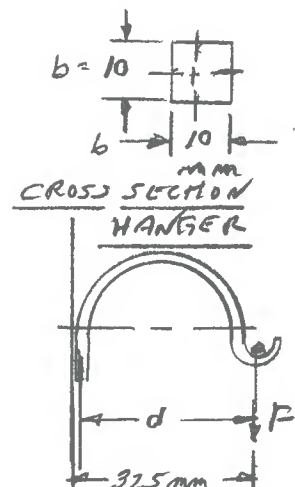
$$M = F \cdot d = F(325 - b/2) = 320F \text{ (NEGATIVE)}$$

$$\sigma_o = \frac{(-320F)(154.946 - 160)}{(100)(160)(155 - 154.946)} = \frac{1.87948F}{\text{mm}^2}$$

$$\sigma_i = \frac{(-320F)(154.946 - 150)}{(100)(150)(155 - 154.946)} = \frac{-1.962F}{\text{mm}^2} \text{ MAXIMUM AT INSIDE SURFACE}$$

$$\text{LET } \sigma_{\text{MAX}} = \frac{-1.962F}{\text{mm}^2} = -248 \text{ N/mm}^2; \quad F = \frac{-248 \text{ N}}{-1.962} = 126.4 \text{ N}$$

COMPRESSIVE YIELD STRENGTH ANSWER



82

CURVED BEAM: COPING SAW FIND N FOR 120 TENSION IN

SAE 1020 CD STEEL; $S_y = 352 \text{ MPa}$

$$M = F \cdot d = (120 \text{ N})(145 \text{ mm}) = 17400 \text{ N} \cdot \text{mm} \text{ (NEG.)}$$

$$r_i = 22 \text{ mm GIVEN}; \quad r_o = 22 + 10 = 32 \text{ mm}$$

$$r_c = 22 + 5 = 27 \text{ mm}$$

$$ASF = b \ln(r_o/r_i) = 4 \ln(32/22) = 1.4987 \text{ mm}$$

$$R = A / ASF = 40 / 1.4987 = 26.688 \text{ mm}$$

SEE PROB. 81 FOR EQUATIONS.

$$(R - r_i) = 26.688 - 22 = 4.688 \text{ mm}$$

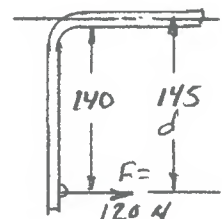
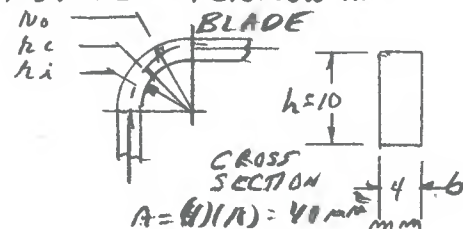
$$(r_c - R) = 27.0 - 26.688 = 0.3115 \text{ mm}$$

$$(R - r_o) = 26.688 - 32.0 = -5.3115 \text{ mm}$$

$$\sigma_i = \frac{(-17400)(4.688)}{(40)(22)(0.3115)} = -297.6 \text{ MPa COMPRESSION ON INSIDE SURFACE}$$

$$\sigma_o = \frac{(-17400)(-5.3115)}{(40)(32)(0.3115)} = 231.8 \text{ MPa TENSION ON OUTSIDE SURFACE}$$

$$\begin{aligned} N_i &= \frac{S_y}{\sigma_i} = \frac{-352 \text{ MPa}}{-297.6 \text{ MPa}} = 1.18 \text{ MINIMUM} \\ N_o &= \frac{S_y}{\sigma_o} = \frac{352 \text{ MPa}}{231.8 \text{ MPa}} = 1.52 \end{aligned} \quad \left. \vphantom{\begin{aligned} N_i &= \frac{S_y}{\sigma_i} \\ N_o &= \frac{S_y}{\sigma_o} \end{aligned}} \right\} \text{ BOTH LOW}$$



83

CURVED BEAM HACK SAW FIG. P3-83

GIVEN: $F = 480 \text{ N}$; FIND N .SAE 120 CD; $S_y = 352 \text{ MPa}$

CONSIDER TUBE TO BE A COMPOSITE SECTION: ① - ②

$$A_1 = \pi(10)^2/4 = 78.54 \text{ mm}^2; A_2 = \pi(6)^2/4 = 28.27 \text{ mm}^2$$

$$A = A_1 - A_2 = 50.265 \text{ mm}^2$$

$$ASF = ASF_1 - ASF_2$$

$$ASF_1 = 2\pi \left[r_c - (r_c^2 - D_1^2/4)^{1/2} \right] = 3.9903 \text{ mm}$$

$$ASF_2 = 2\pi \left[r_c - (r_c^2 - D_2^2/4)^{1/2} \right] = 1.4218 \text{ mm}$$

$$ASF = 3.9903 - 1.4218 = 2.5685 \text{ mm}$$

$$R = \frac{A}{ASF} = \frac{50.265 \text{ mm}^2}{2.5685 \text{ mm}} = 19.569 \text{ mm}$$

SEE PROBLEM B1 FOR EQUATIONS:

$$(R - r_o) = (19.569 - 25) = -5.4307 \text{ mm}$$

$$(r_c - R) = (20 - 19.569) = 0.4307 \text{ mm}$$

$$(R - r_i) = (19.569 - 15) = 4.569 \text{ mm}$$

$$M = F \cdot d = 480 \text{ N}(80 \text{ mm}) = 38400 \text{ N} \cdot \text{mm}$$

$$\sigma_i = \frac{(-38400)(4.569)}{(50.265)(15)(0.4307)} = -540.3 \text{ MPa COMPRESSION ON INSIDE SURFACE}$$

$$\sigma_o = \frac{(-38400)(-5.4307)}{(50.265)(25)(0.4307)} = 385.3 \text{ MPa TENSION ON OUTSIDE SURFACE}$$

$$\left. \begin{aligned} N_i &= \frac{S_y}{\sigma_i} = \frac{-352 \text{ MPa}}{-540.3 \text{ MPa}} = 0.651 \\ N_o &= \frac{S_y}{\sigma_o} = \frac{352 \text{ MPa}}{385.3 \text{ MPa}} = 0.913 \end{aligned} \right\} \text{ BOTH INDICATE FAILURE}$$

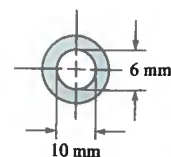
ADDITION TO PROBLEM: FIND F_{ALL} TO ACHIEVE $N \geq 2.0$ STRESSES ARE PROPORTIONAL TO MOMENT AND TO F .

$$\text{STRESS REDUCTION REQD: } \sigma_{ALL} = \frac{S_y}{2} = \frac{352 \text{ MPa}}{2} = 176 \text{ MPa}$$

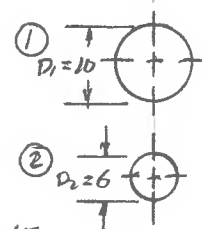
$$\frac{\sigma_{ALL}}{\sigma_i} = \frac{-176 \text{ MPa}}{-540.3 \text{ MPa}} = 0.326 = \frac{M_{ALL}}{M_1}$$

$$M_{ALL} = 0.326 M_1 = 0.326(-38400) = -12509 \text{ N} \cdot \text{mm} = F_{ALL} \cdot d$$

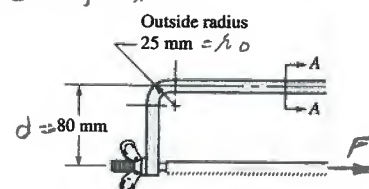
$$F_{ALL} = \frac{12509 \text{ N} \cdot \text{mm}}{80 \text{ mm}} = 156.4 \text{ N}$$



Section A-A



$$r_c = 20; r_i = 15$$



84

CURVED BEAM GARDEN TOOLFIND F FOR YIELDING.CAST ALUMINUM: 356.0-T6, $S_y = 207 \text{ MPa}$

$$ASF = 2\pi \left[r_c - (r_c^2 - D^2/4)^{1/2} \right]$$

$$ASF = 2\pi \left[12 - (12^2 - 8^2/4)^{1/2} \right] = 4,312 \text{ mm}$$

$$R = \frac{A}{ASF} = \frac{50.265 \text{ mm}^2}{4,312 \text{ mm}} = 11.6568 \text{ mm}$$

$$(r_c - R) = (12 - 11.6568) = 0.3431 \text{ mm}$$

$$(R - r_o) = (11.6568 - 16) = -4.343 \text{ mm}$$

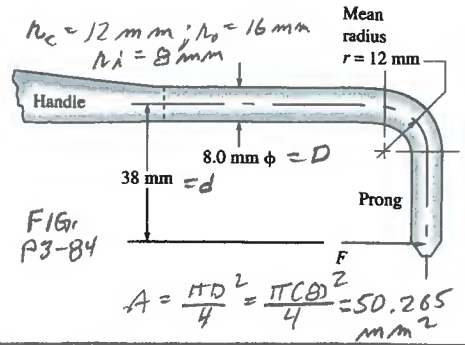
$$(R - r_i) = (11.6568 - 8) = 3.6568 \text{ mm}$$

LET $M = F \cdot d$, $\sigma = S_y = 207 \text{ MPa}$ (POS.)

$$F_{\text{AIL}} = \frac{(-207 \text{ N/mm}^2)(50.265 \text{ mm}^2)(16 \text{ mm})(0.3431 \text{ mm})}{(38 \text{ mm})(-4.343 \text{ mm})} = 346.1 \text{ N}$$

SIMILARLY:

$$F_{\text{AIL}} = \frac{S_y A r_i (r_c - R)}{d (R - r_i)} = \frac{(207)(50.265)(8)(0.3431)}{(38)(3.6568)} = 205.6 \text{ N}$$

GOVERNS

$$A = \frac{\pi D^2}{4} = \frac{\pi (8)^2}{4} = 50.265 \text{ mm}^2$$

$$\sigma_o = S_y = \frac{(E \cdot d) (R - r_o)}{A r_o (r_c - R)}$$

$$F_{\text{AIL}} = \frac{S_y (A) r_o (r_c - R)}{d (R - r_o)}$$

LET $S_y = -207 \text{ MPa}$ COMPRESSION

85

CURVED BEAM HOOP SUPPORT FIG. P3-85: $r_i = r_o - D_i$ STEEL: ASTM A53-GR B $S_y = 35 \text{ KSI}$

$$M = F \cdot d = (230 \text{ lb})(48 \text{ in}) = -11040 \text{ lb-in (NEG.)}$$

ANALYSIS AS IN PROB 3-83

STRESS EQNS. IN PROB 3-81

$$ASF = ASF_1 - ASF_2 = 0.61748 - 0.45484 = 0.16264 \text{ in}$$

$$R = \frac{A}{ASF} = \frac{1.704 \text{ in}^2}{0.16264 \text{ in}} = 10.4769 \text{ in}$$

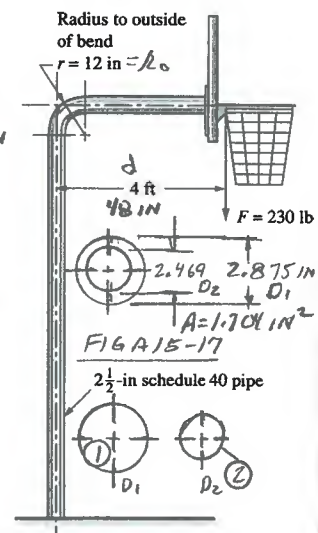
$$(R - r_o) = -1.5231 \text{ in}; (R - r_i) = 1.35185 \text{ in}; (r_c - R) = 0.08564 \text{ in}$$

$$\sigma_i = \frac{(-11040 \text{ lb-in})(1.35185 \text{ in})}{(1.704 \text{ in}^2)(9.125 \text{ in})(0.08564 \text{ in})} = -11208 \text{ psi COMP.}$$

$$\sigma_o = \frac{(-11040 \text{ lb-in})(-1.5231)}{(1.704 \text{ in}^2)(12)(0.08564 \text{ in})} = 9602.4 \text{ psi TENSION}$$

$$N_i = \frac{S_y}{\sigma_i} = \frac{-35000 \text{ psi}}{-11208 \text{ psi}} = 3.123 \text{ LOWEST - GOVERNS}$$

$$N_o = \frac{S_y}{\sigma_o} = \frac{35000 \text{ psi}}{9602.4 \text{ psi}} = 3.645$$

SATISFACTORY N 

86

CURVED BEAM C-CLAMP FIG. P3-86

CAST ZINC ZA-12; $S_{UT} = 404 \text{ MPa}$
 $S_{UC} = 269 \text{ MPa}$

FIND: F FOR $N=3$

$$\bar{y} = \frac{1}{A} \left[(b_1 f_1) \bar{y}_1 + (b_2 f_2) (\bar{y}_1 + \bar{y}_2) \right]$$

$$\bar{y} = \frac{1}{57} \left[(8 \cdot 3)(1.5) + (3 \cdot 11)(3 + 5.5) \right] = 5.553 \text{ mm}$$

$$ASF = b_1 \ln \left(\frac{r_1}{r_2} \right) + b_2 \ln \left(\frac{r_2}{r_1} \right); r_1 = r_i + f_1 = 5 + 3 = 8 \text{ mm}$$

$$ASF = 8 \ln \left(\frac{8}{5} \right) + 3 \ln \left(\frac{19}{8} \right) = 6.355 \text{ mm}$$

$$R = A/ASF = 57/6.355 = 8.969 \text{ mm}$$

$$M = F(26 + \bar{y}) = F(26 + 5.553) = 31.553 F \text{ (POS.)}$$

$$\sigma_i = \frac{M(R - r_i)}{A r_i (r_o - R)} = \frac{31.553 F(8.969)}{(57)(5)(1.583)} = 0.2776 F_i \text{ TENSION}$$

$$\text{LET } \sigma_i = \frac{S_{UT}}{3} = \frac{404 \text{ MPa}}{3} = 134.67 \text{ MPa}$$

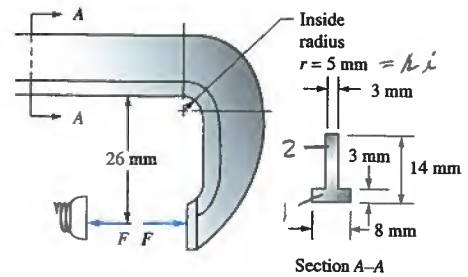
$$F_i = \frac{134.67 \text{ N/mm}^2}{0.2776 \text{ mm}^2} = 485.1 \text{ N}$$

$$\sigma_o = \frac{M(R - r_o)}{A r_o (r_o - R)} = \frac{31.553 F(-10.0307)}{(57)(19)(1.583)} = -0.1846 F_o \text{ COMPRESSION}$$

$$\text{LET } \sigma_o = \frac{S_{UC}}{3} = \frac{-269 \text{ MPa}}{3} = -89.67 \text{ N/mm}^2$$

$$F_o = \frac{-89.67 \text{ N/mm}^2}{-0.1846 \text{ mm}^2} = 485.7 \text{ N}$$

BECAUSE $F_o \approx F_i$, THE DESIGN OF THE INVERTED T-SECTION USES THE MATERIAL VERY EFFICIENTLY.



$$\text{PART 1: } A_1 = 3 \cdot 8 = 24 \text{ mm}^2$$

$$\text{PART 2: } A_2 = 3 \cdot 11 = 33 \text{ mm}^2$$

$$A = A_1 + A_2 = 57 \text{ mm}^2$$

$$r_o = r_i + 14 = 5 + 14 = 19 \text{ mm}$$

$$r_c = r_i + \bar{y} = 5 + 5.553$$

$$r_c = 10.553 \text{ mm}$$

$$(R - r_i) = 8.969 - 5$$

$$= 3.969 \text{ mm}$$

$$(R - r_o) = 8.969 - 19$$

$$= -10.0307 \text{ mm}$$

$$(r_c - R) = 10.553 - 8.969$$

$$= 1.583 \text{ mm}$$

CHAPTER 4 COMBINED STRESSES AND MOHR'S CIRCLE

NOTE: The solutions to Chapter 4 problems 1 – 30 are shown on the following pages as images of the output from the MDESIGN – MOTT software that is included in the text. Each problem produces a solution in line with the procedure shown for manual solution in Section 4-4 in Chapter 4 of the text and as shown in the four Example Problems in Section 4-5 of the text. Problem 4-1 is shown worked out in manual form below and the MDESIGN – MOTT solution is shown on the following page. Solutions for all other problems are shown only as the results from the MDESIGN – MOTT solutions. Note that in the MDESIGN-MOTT output, the graphic view of Mohr's circle and the stress elements show the stress values only in psi.

1

$$\sigma_x = 20 \text{ ksi}; \sigma_y = 0; \tau_{xy} = 10 \text{ ksi}; \text{ X-AXIS IN 1ST QUADRANT}$$

$$\sigma_{\text{AVG}} = (\sigma_x + \sigma_y)/2 = (20 + 0)/2 = 10.0 \text{ ksi}$$

$$a = \sigma_x - \sigma_{\text{AVG}} = 20 - 10 = 10 \text{ ksi}; b = \tau_{xy} = 10 \text{ ksi}$$

$$R = \sqrt{a^2 + b^2} = \sqrt{10^2 + 10^2} = 14.14 \text{ ksi} = \tau_{\text{MAX}}$$

$$\alpha = \tan^{-1}(b/a) = \tan^{-1}(10/10) = 45^\circ = 2\phi_\sigma \text{ CW FROM X-AXIS TO } \sigma_1$$

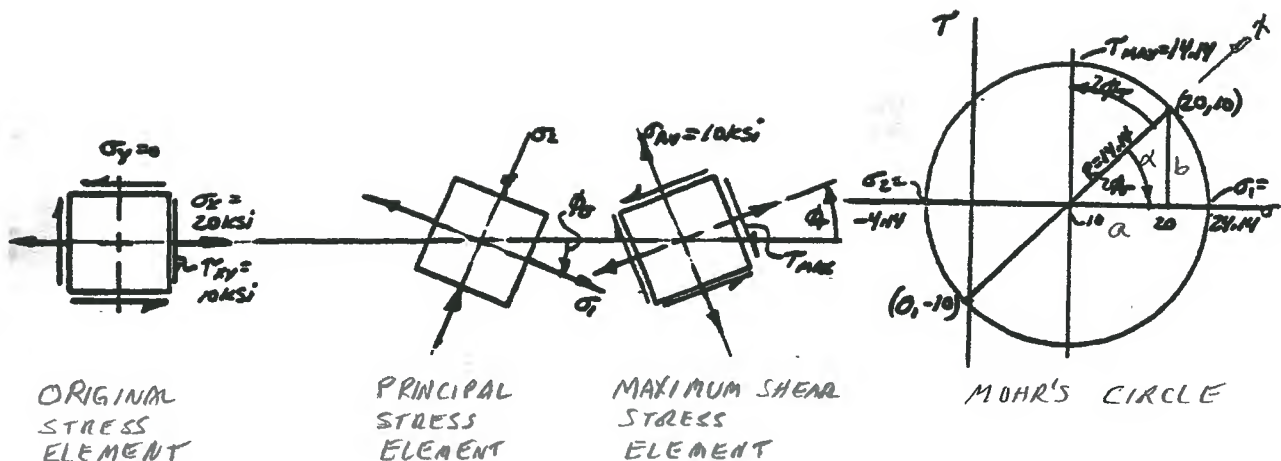
$$\phi_\sigma = 2\phi_\sigma/2 = 45.0^\circ/2 = 22.5^\circ$$

$$2\phi_\tau = 90^\circ - \alpha = 90 - 45 = 45^\circ \text{ CCW FROM X-AXIS TO } \tau_{\text{MAX}}$$

$$\phi_\tau = 2\phi_\tau/2 = 45/2 = 22.5^\circ$$

$$\sigma_1 = \sigma_{\text{AVG}} + R = 10 + 14.14 = 24.14 \text{ ksi}$$

$$\sigma_2 = \sigma_{\text{AVG}} - R = 10 - 14.14 = -4.14 \text{ ksi}$$



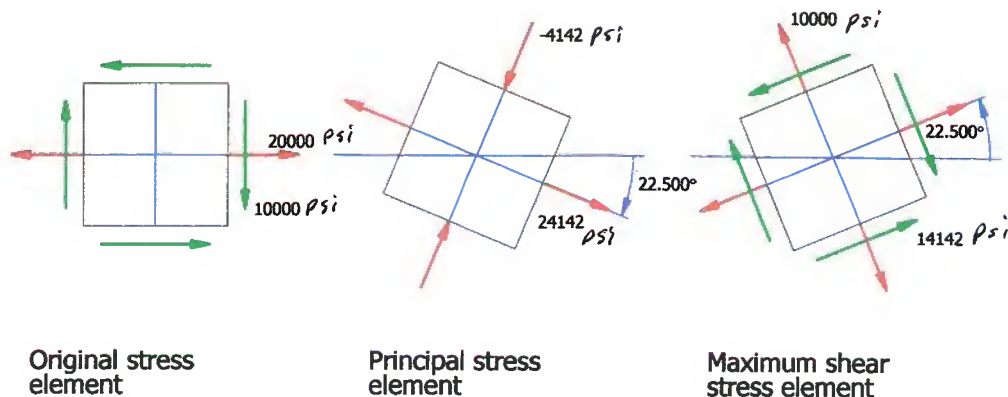
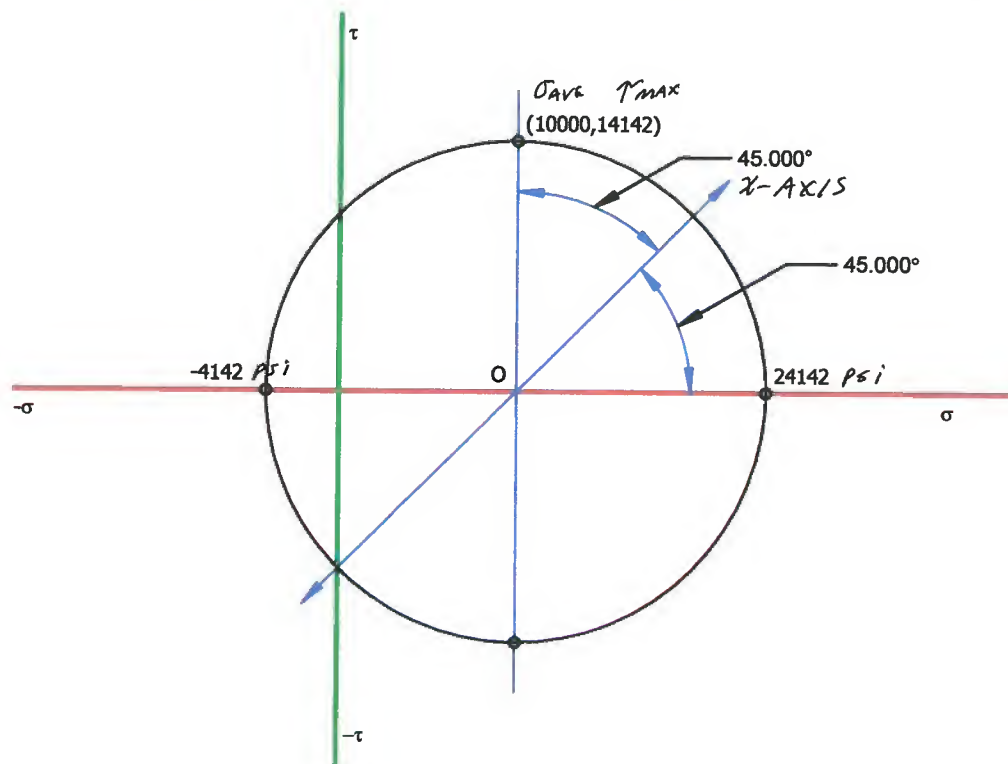


$\sigma_x = 20$ ksi
 $\sigma_y = 0$ ksi
 $\tau_{xy} = 10$ ksi

Results:

Maximum principal stress	$\sigma_1 =$	24.142	ksi
Minimum principal stress	$\sigma_2 =$	-4.142	ksi
Maximum shear stress	$\tau_{max} =$	14.142	ksi
Average normal stress	$\sigma_{avg} =$	10.000	ksi
Principal planes	$\phi_\sigma =$	22.500	°
Angle of maximum shear stress	$\phi_\tau =$	22.500	°

CW
CCW



2

 $\sigma_x = -85000$ psi $\sigma_y = 40000$ psi $\tau_{xy} = 30000$ psi**Results:**

Maximum principal stress

 $\sigma_1 = 46827.123$ psi

Minimum principal stress

 $\sigma_2 = -91827.123$ psi

Maximum shear stress

 $\tau_{max} = 69327.123$ psi

Average normal stress

 $\sigma_{avg} = -22500.000$ psi

Principal planes

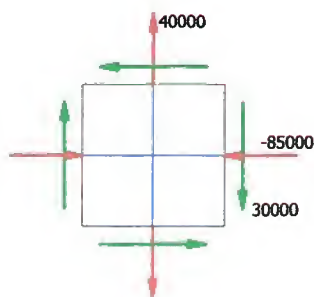
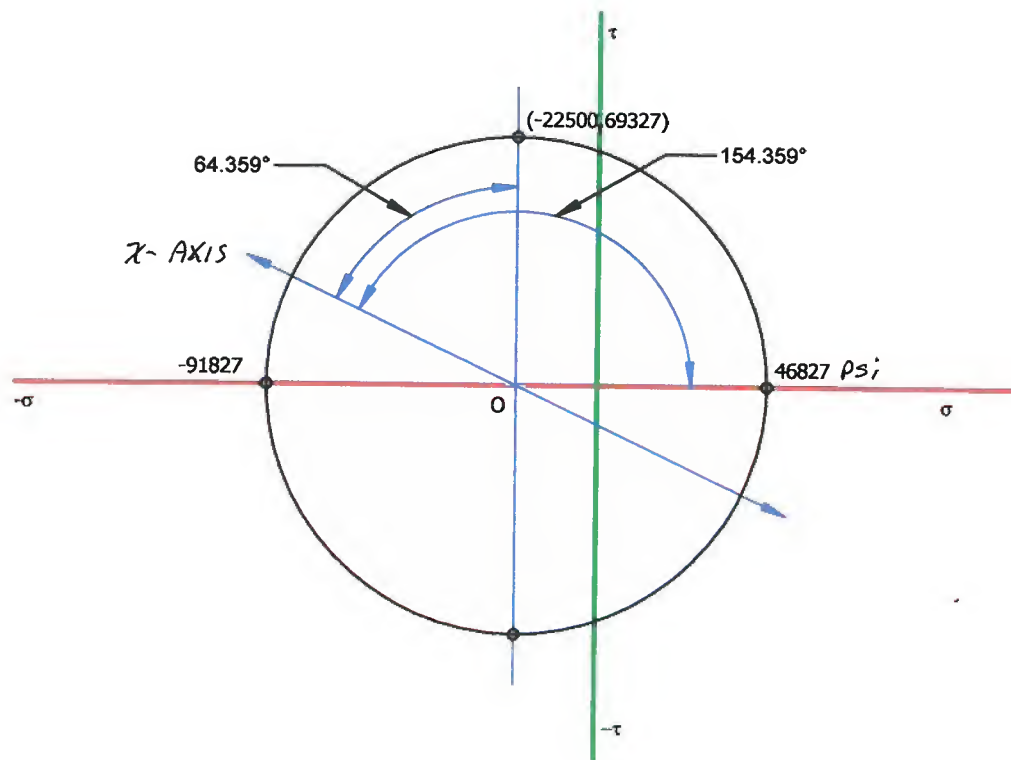
 $\phi_\sigma = 77.179^\circ$

Angle of maximum shear stress

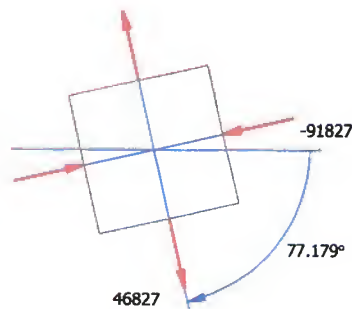
 $\phi_\tau = 32.179^\circ$

CW

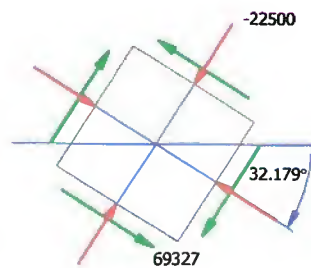
CW



Original stress element



Principal stress element



Maximum shear stress element

3

$\sigma_x = 40$ ksi
 $\sigma_y = -40$ ksi
 $\tau_{xy} = -30$ ksi

Results:

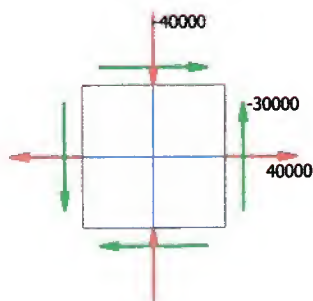
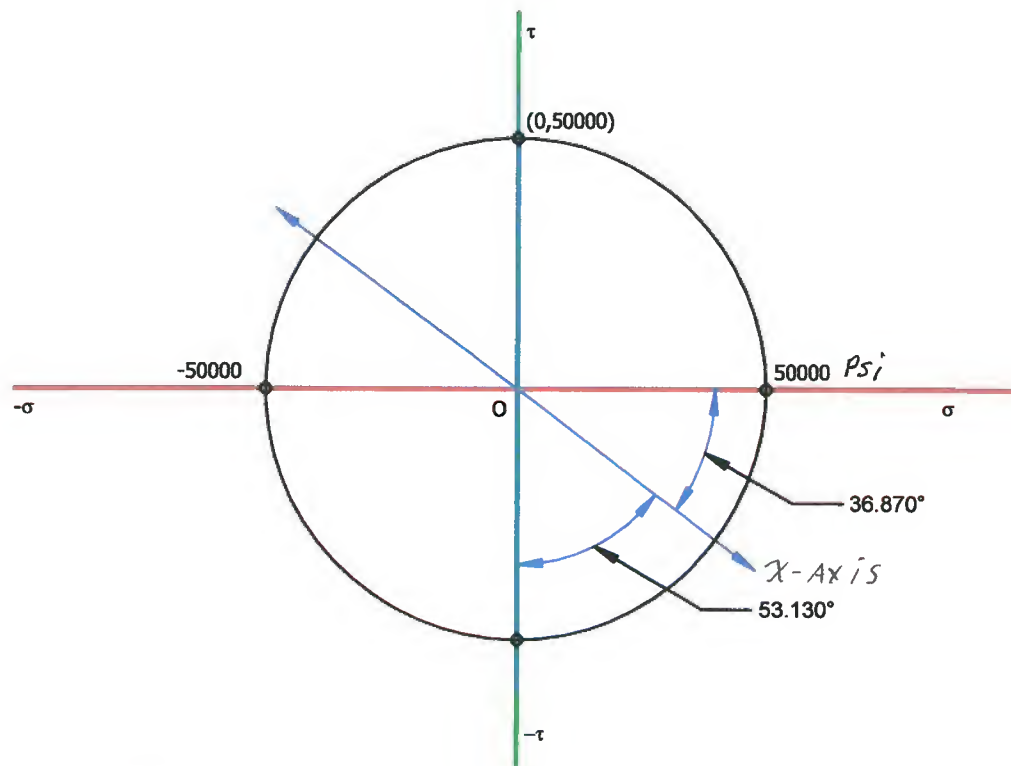
Maximum principal stress $\sigma_1 = 50.000$ ksi
 Minimum principal stress $\sigma_2 = -50.000$ ksi
 Maximum shear stress $\tau_{max} = 50.000$ ksi
 Average normal stress $\sigma_{avg} = 0.000$ ksi
 Principal planes $\phi_\sigma = 18.435^\circ$

Angle of maximum shear stress

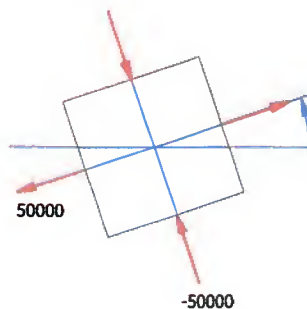
$\phi_\tau = 26.565^\circ$

CCW

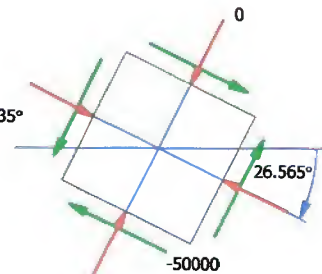
cw to $-\tau_{max}$



Original stress element



Principal stress element



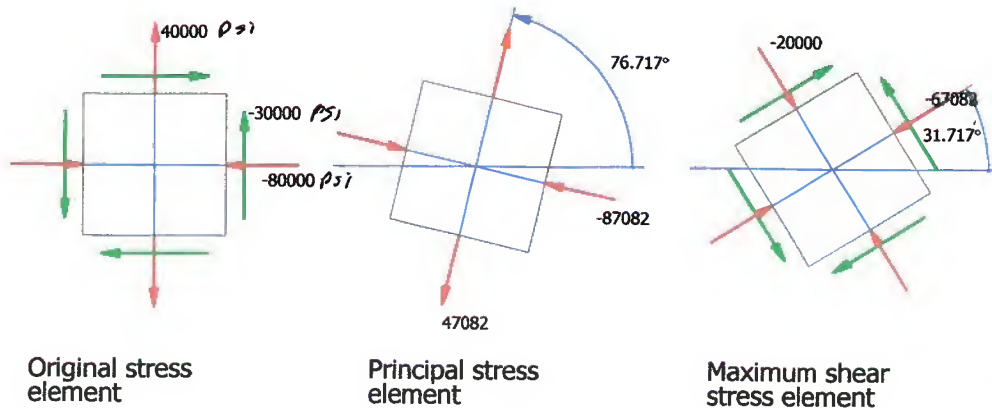
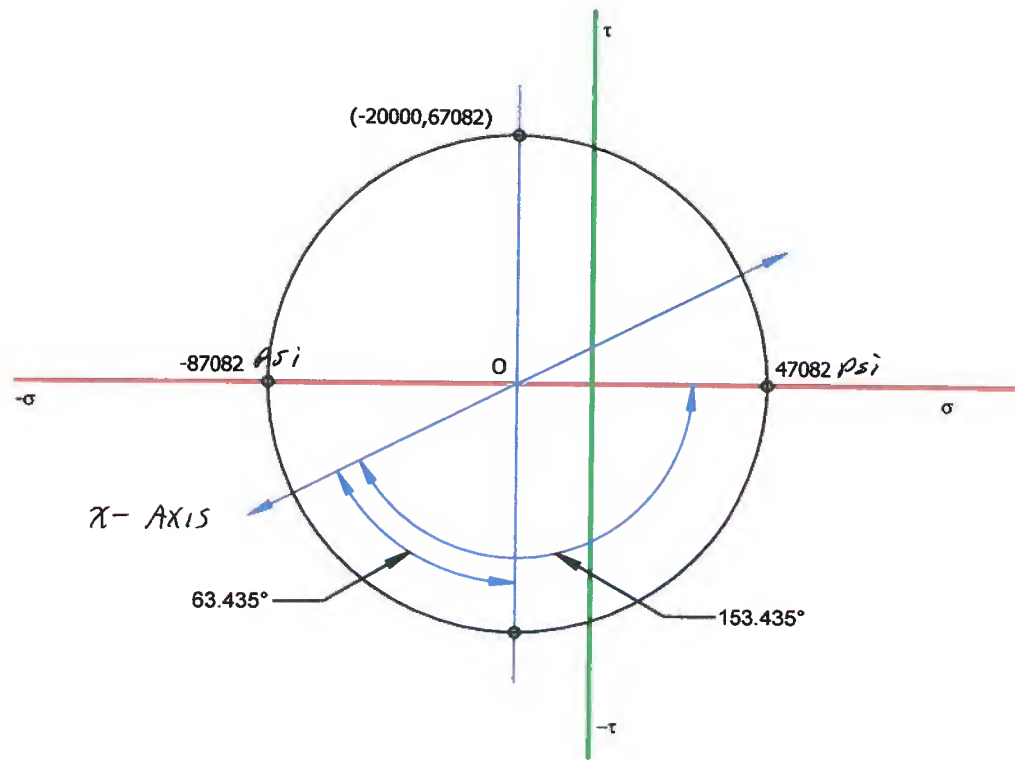
Maximum shear stress element

4

$\sigma_x = -80$ ksi
 $\sigma_y = 40$ ksi
 $\tau_{xy} = -30$ ksi

Results:

Maximum principal stress	$\sigma_1 =$	47.082	ksi
Minimum principal stress	$\sigma_2 =$	-87.082	ksi
Maximum shear stress	$\tau_{max} =$	67.082	ksi
Average normal stress	$\sigma_{avg} =$	-20.000	ksi
Principal planes	$\phi_\sigma =$	76.717	°
			ccw
Angle of maximum shear stress	$\phi_\tau =$	31.717	°
			ccw to $-\tau_{max}$

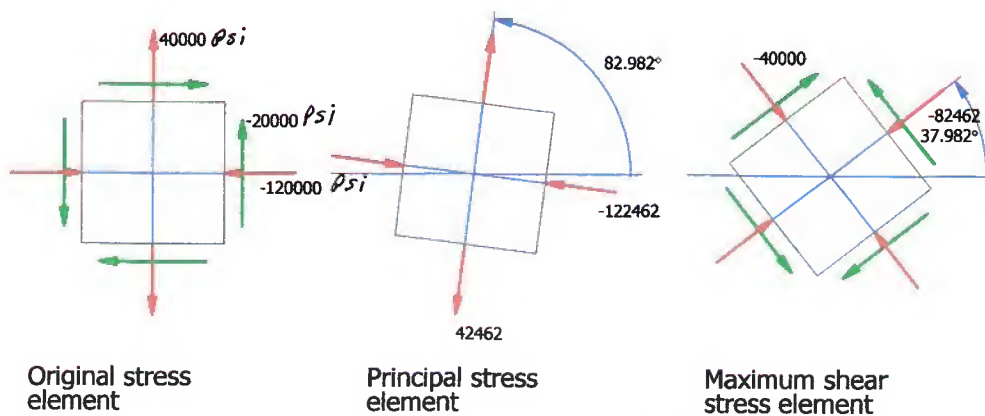
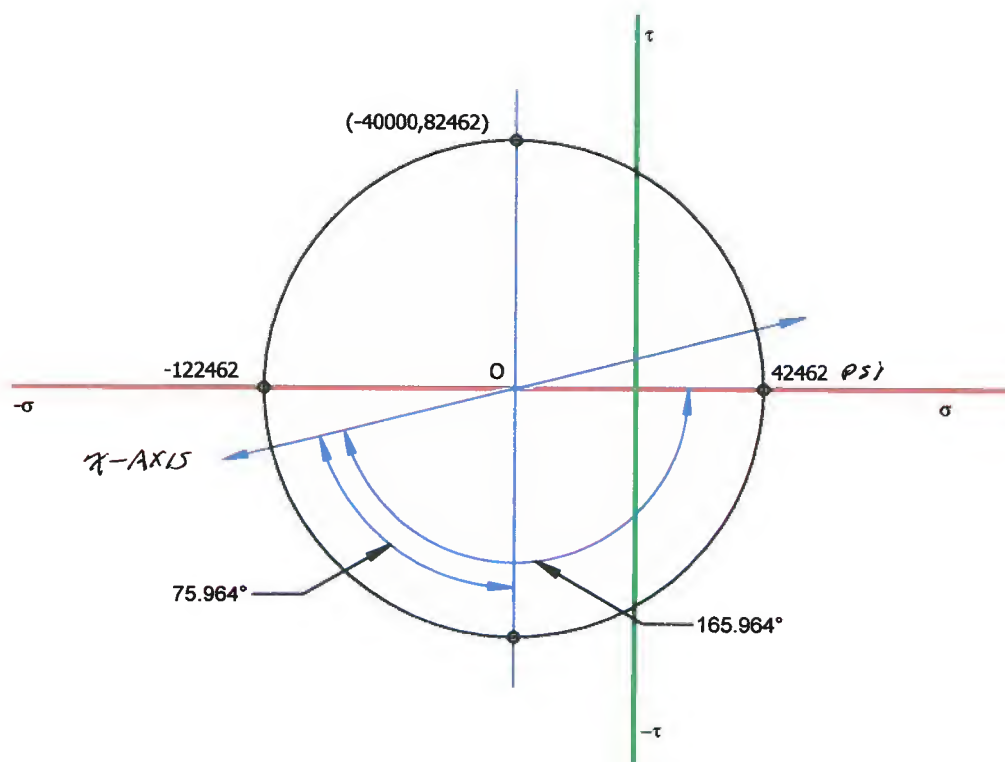


5

$$\begin{aligned}\sigma_x &= -120000 \text{ psi} \\ \sigma_y &= 40000 \text{ psi} \\ \tau_{xy} &= -20000 \text{ psi}\end{aligned}$$

Results:

Maximum principal stress	$\sigma_1 = 42462.113$	psi
Minimum principal stress	$\sigma_2 = -122462.113$	psi
Maximum shear stress	$\tau_{\max} = 82462.113$	psi
Average normal stress	$\sigma_{\text{avg}} = -40000.000$	psi
Principal planes	$\phi_\sigma = 82.982$	°
		CCW
Angle of maximum shear stress	$\phi_\tau = 37.982$	°
		ccw to $-\tau_{\max}$



6

$$\sigma_x = -120 \quad \text{ksi}$$

$$\sigma_y = 40 \quad \text{ksi}$$

$$\tau_{xy} = 20 \quad \text{ksi}$$

Results:

$$\text{Maximum principal stress} \quad \sigma_1 = 42.462 \quad \text{ksi}$$

$$\text{Minimum principal stress} \quad \sigma_2 = -122.462 \quad \text{ksi}$$

$$\text{Maximum shear stress} \quad \tau_{\max} = 82.462 \quad \text{ksi}$$

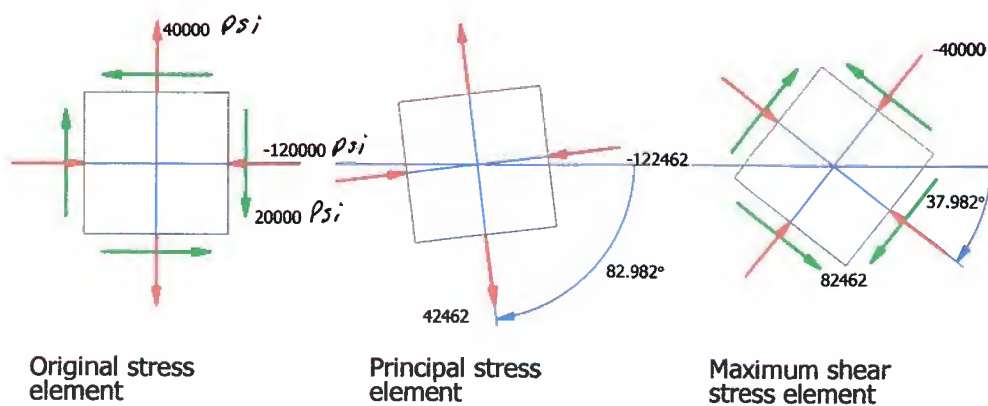
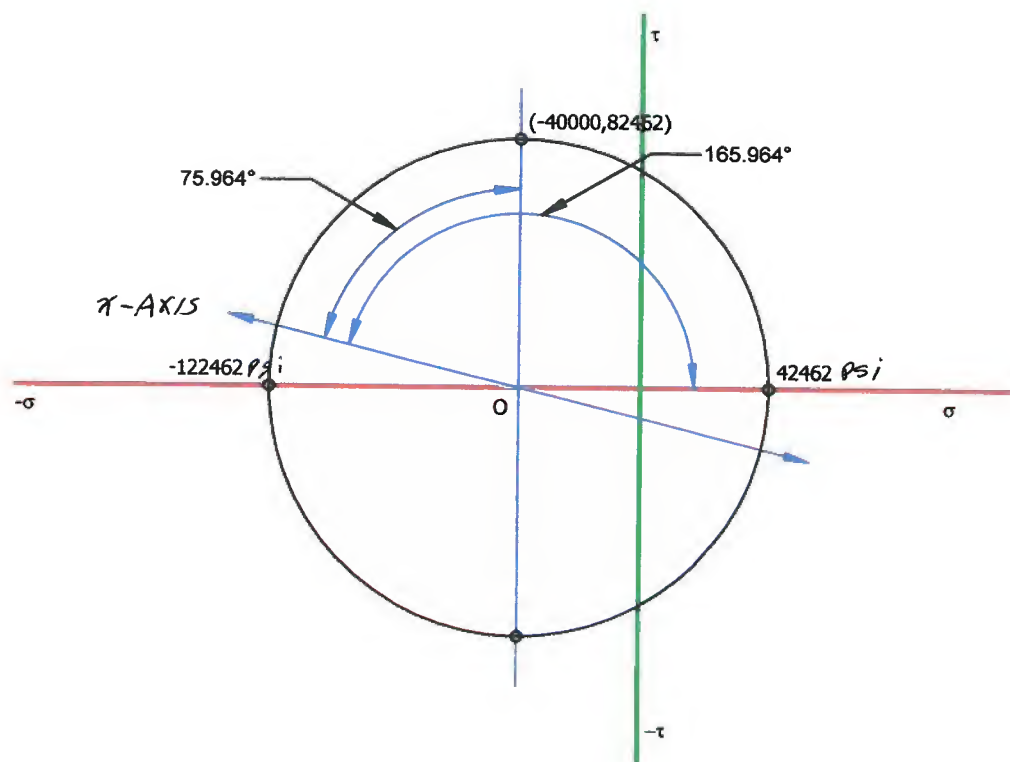
$$\text{Average normal stress} \quad \sigma_{\text{avg}} = -40.000 \quad \text{ksi}$$

$$\text{Principal planes} \quad \phi_{\sigma} = 82.982 \quad ^{\circ}$$

$$\text{Angle of maximum shear stress} \quad \phi_{\tau} = 37.982 \quad ^{\circ}$$

CW

CW



$$\begin{aligned}\sigma_x &= 60000 && \text{psi} \\ \sigma_y &= -40000 && \text{psi} \\ \tau_{xy} &= -35000 && \text{psi}\end{aligned}$$

Maximum principal stress

$$\sigma_1 = 71032.778 \text{ psi}$$

Minimum principal stress

$$\sigma^2 = -51032.778 \text{ psi}$$

Maximum shear stress

$\tau_{max} = 61032.778$ psi

Average normal stress

$$\sigma_{avg} = 10000.000 \text{ psi}$$

Principal planes

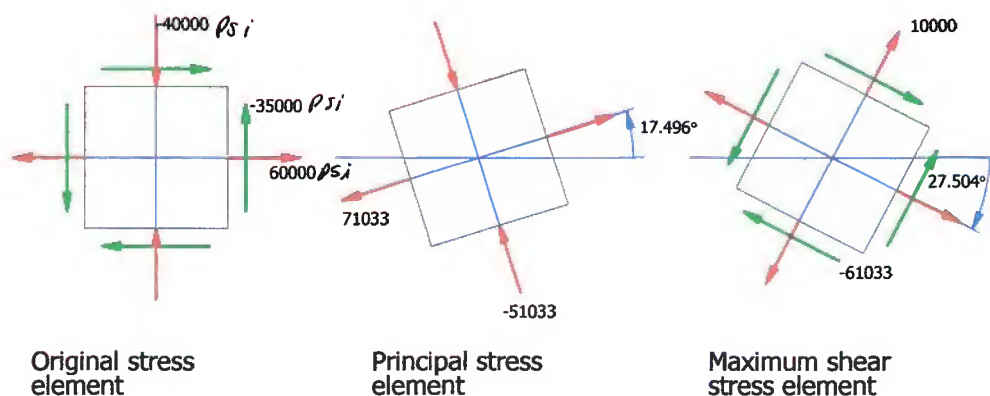
$$\phi\sigma = 17.496$$

Angle of maximum shear stress

$$\phi\tau = 27.504^\circ$$

CCW

cw to $-\tau_{\max}$

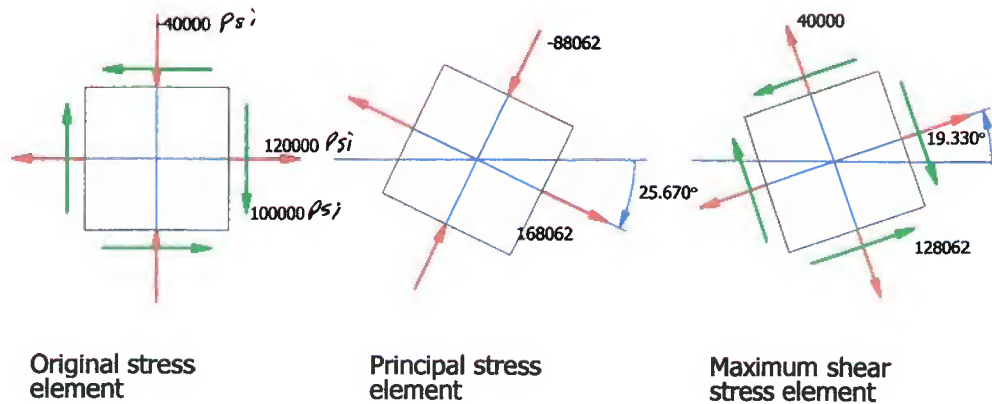
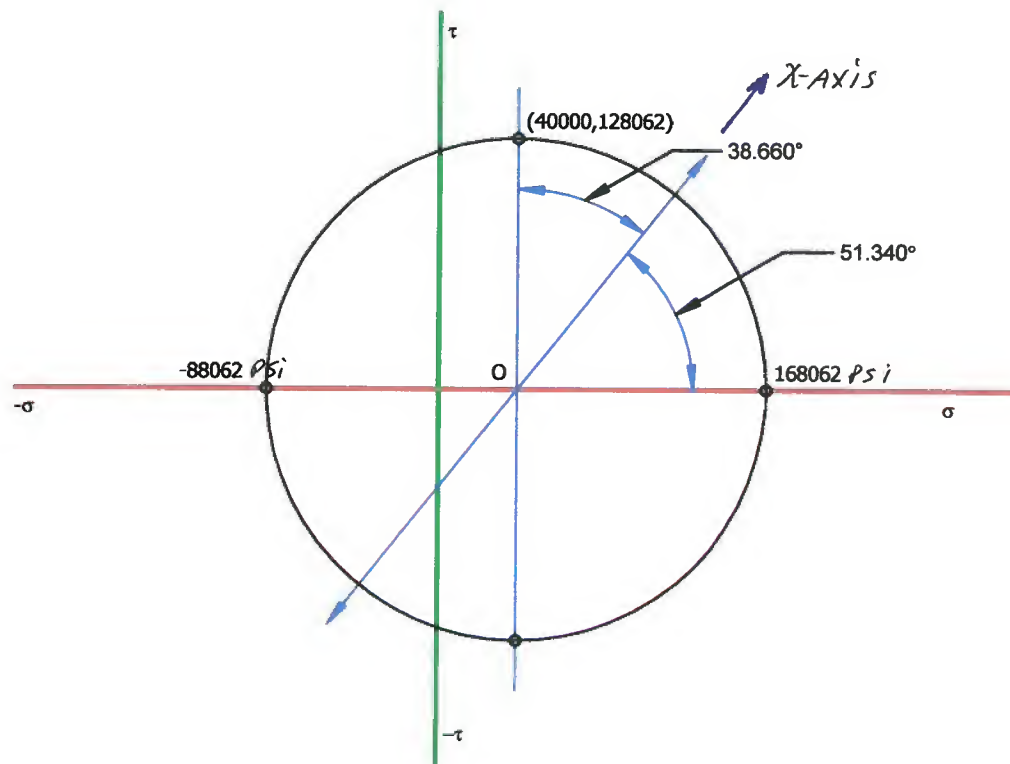


8

$$\begin{aligned}\sigma_x &= 120 && \text{ksi} \\ \sigma_y &= -40 && \text{ksi} \\ \tau_{xy} &= 100 && \text{ksi}\end{aligned}$$

Results:

Maximum principal stress	$\sigma_1 =$	168.062	ksi
Minimum principal stress	$\sigma_2 =$	-88.062	ksi
Maximum shear stress	$\tau_{\max} =$	128.062	ksi
Average normal stress	$\sigma_{\text{avg}} =$	40.000	ksi
Principal planes	$\phi_\sigma =$	25.670	°
			CW
Angle of maximum shear stress	$\phi_\tau =$	19.330	°
			CCW

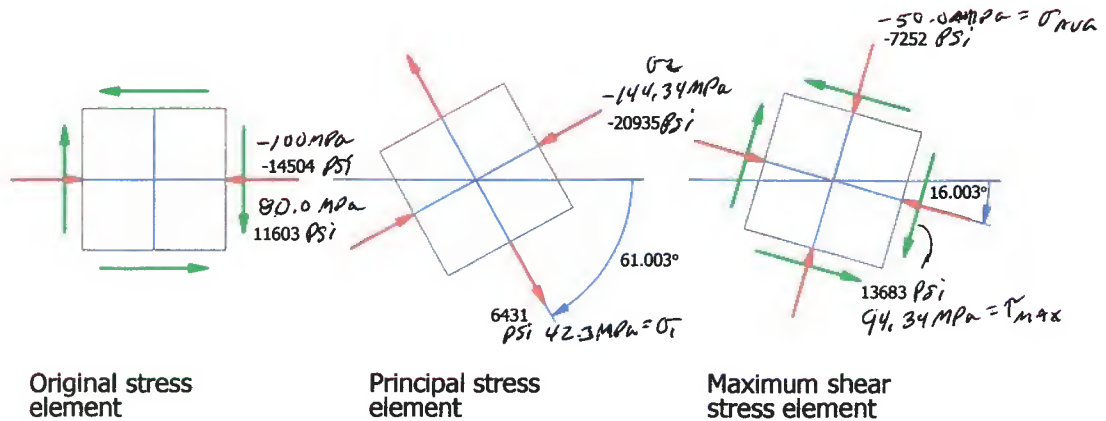
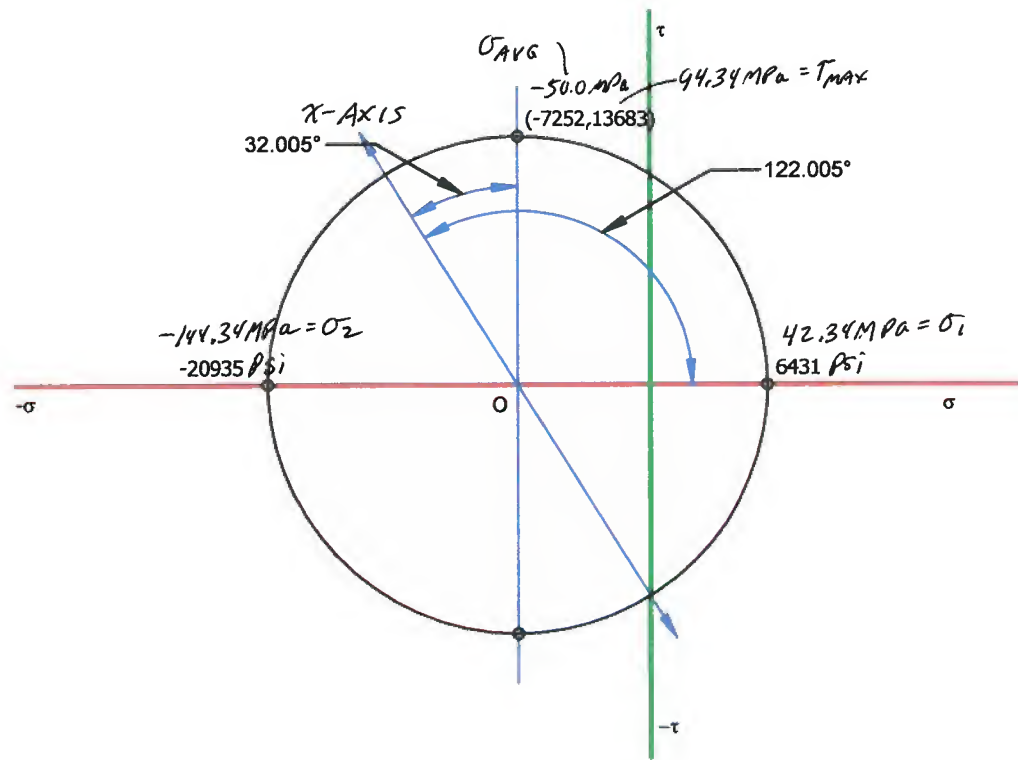


9

$\sigma_x = -100$ MPa
 $\sigma_y = 0$ MPa
 $\tau_{xy} = 80$ MPa

Results:

Maximum principal stress	$\sigma_1 =$	44.340	MPa
Minimum principal stress	$\sigma_2 =$	-144.340	MPa
Maximum shear stress	$\tau_{max} =$	94.340	MPa
Average normal stress	$\sigma_{avg} =$	-50.000	MPa
Principal planes	$\phi_\sigma =$	61.003	°
			CW
Angle of maximum shear stress	$\phi_\tau =$	16.003	°
			CW

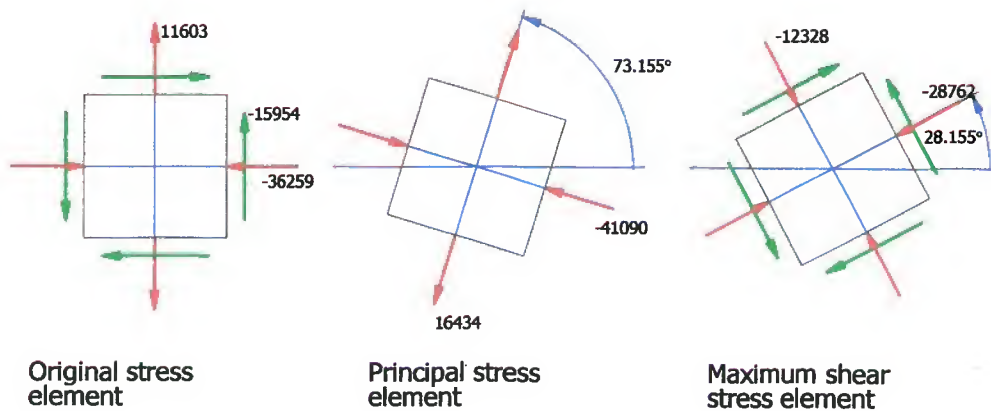
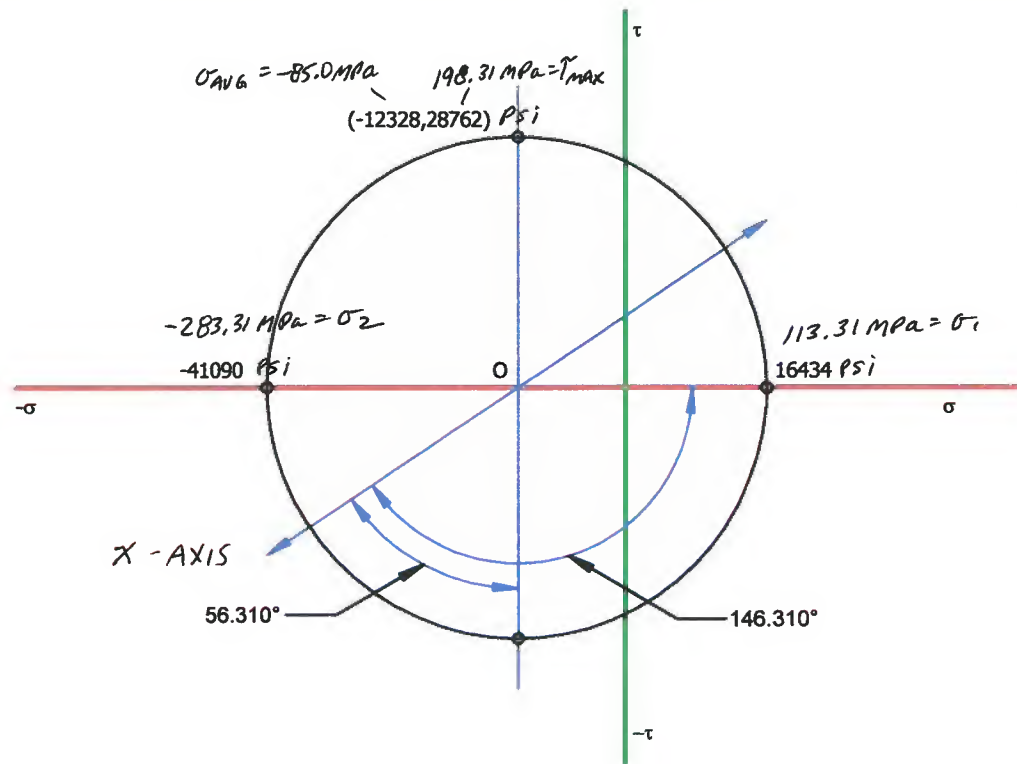


10

$\sigma_x = -250$ MPa
 $\sigma_y = 80$ MPa
 $\tau_{xy} = -110$ MPa

Results:

Maximum principal stress	$\sigma_1 =$	113.305	MPa
Minimum principal stress	$\sigma_2 =$	-283.305	MPa
Maximum shear stress	$\tau_{max} =$	198.305	MPa
Average normal stress	$\sigma_{avg} =$	-85.000	MPa
Principal planes	$\phi_\sigma =$	73.155	°
			CCW
Angle of maximum shear stress	$\phi_\tau =$	28.155	°
			CCW to $-\tau_{max}$



//

$\sigma_x = 50$ MPa
 $\sigma_y = -80$ MPa
 $\tau_{xy} = 40$ MPa

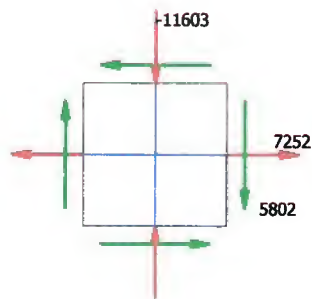
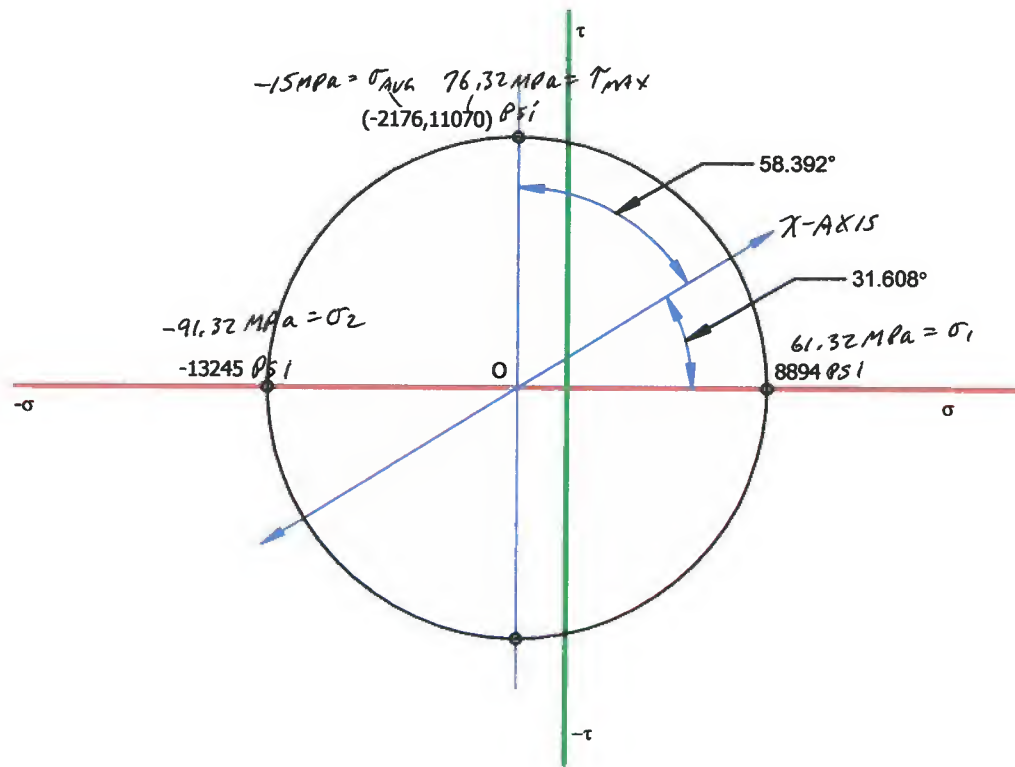
Results:

Maximum principal stress $\sigma_1 = 61.322$ MPa
 Minimum principal stress $\sigma_2 = -91.322$ MPa
 Maximum shear stress $\tau_{max} = 76.322$ MPa
 Average normal stress $\sigma_{avg} = -15.000$ MPa
 Principal planes $\phi_\sigma = 15.804^\circ$

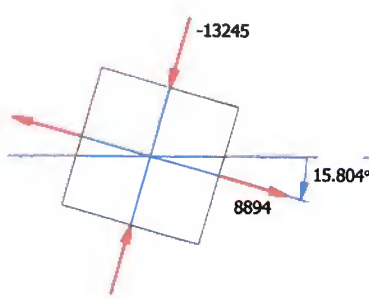
Angle of maximum shear stress

$\phi_\tau = 29.196^\circ$

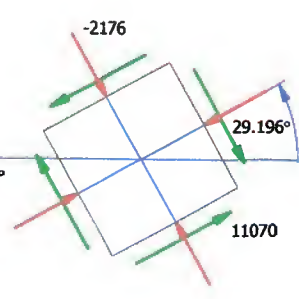
CW
 CCW



Original stress element



Principal stress element



Maximum shear stress element

12

$\sigma_x = 150$ MPa
 $\sigma_y = -80$ MPa
 $\tau_{xy} = -40$ MPa

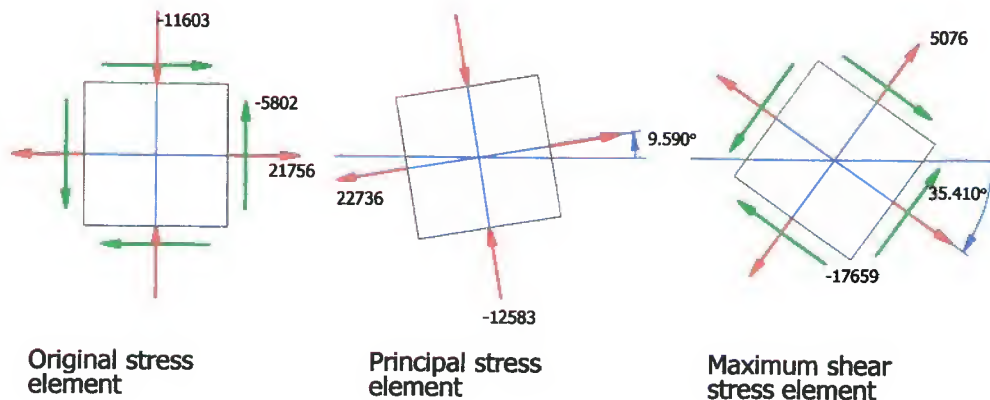
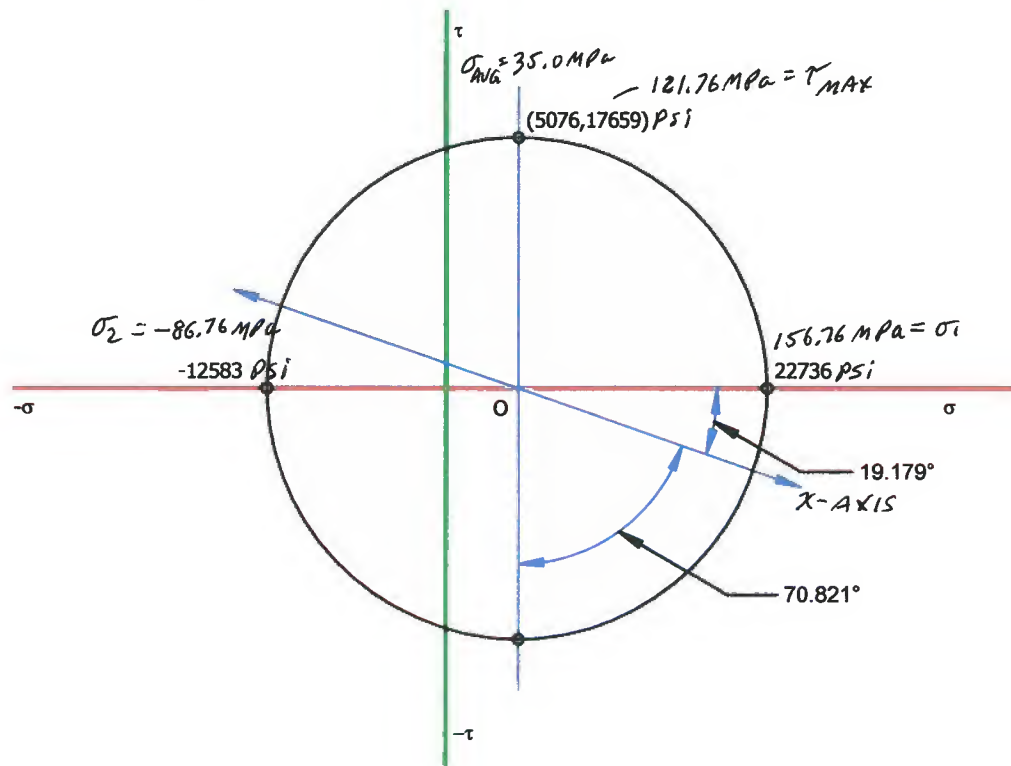
Results:

Maximum principal stress $\sigma_1 = 156.758$ MPa
 Minimum principal stress $\sigma_2 = -86.758$ MPa
 Maximum shear stress $\tau_{max} = 121.758$ MPa
 Average normal stress $\sigma_{avg} = 35.000$ MPa
 Principal planes $\phi_\sigma = 9.590^\circ$

Angle of maximum shear stress

$\phi_\tau = 35.410^\circ$

CCW

CW to $-\tau_{max}$ 

13

$$\sigma_x = -150 \quad \text{MPa}$$

$$\sigma_y = 80 \quad \text{MPa}$$

$$\text{Results: } \tau_{xy} = -40 \quad \text{MPa}$$

Maximum principal stress

$$\sigma_1 = 86.758 \quad \text{MPa}$$

Minimum principal stress

$$\sigma_2 = -156.758 \quad \text{MPa}$$

Maximum shear stress

$$\tau_{\max} = 121.758 \quad \text{MPa}$$

Average normal stress

$$\sigma_{\text{avg}} = -35.000 \quad \text{MPa}$$

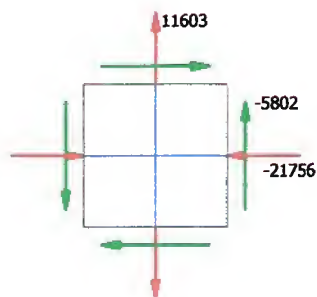
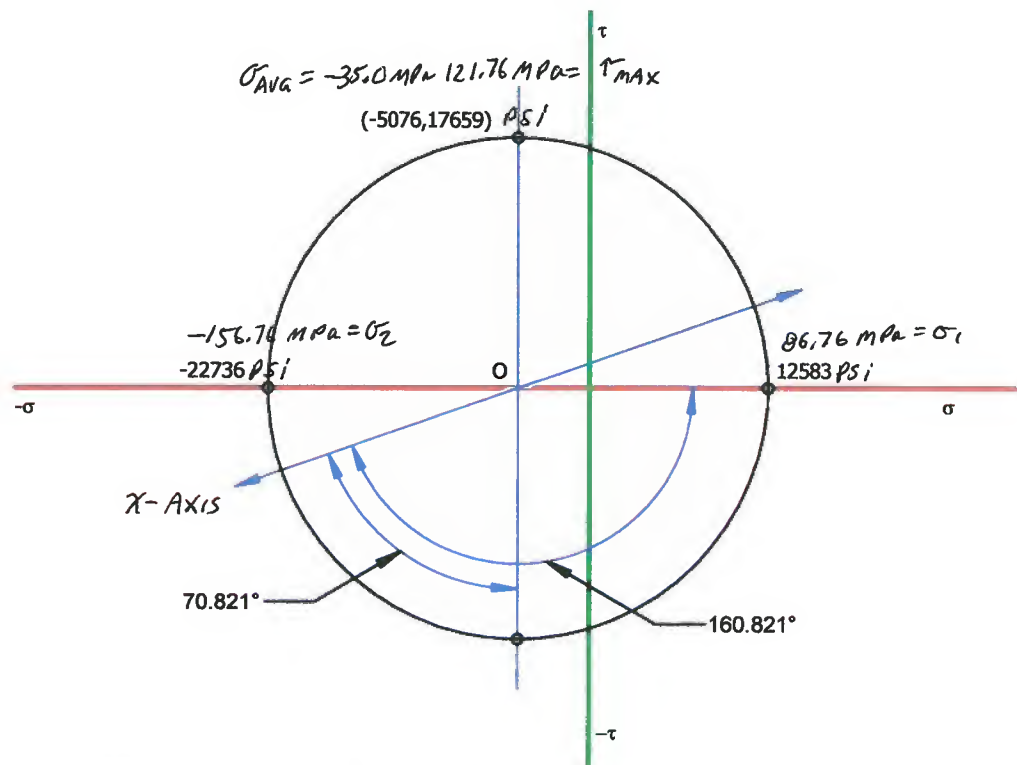
Principal planes

$$\phi_{\sigma} = 80.410^\circ$$

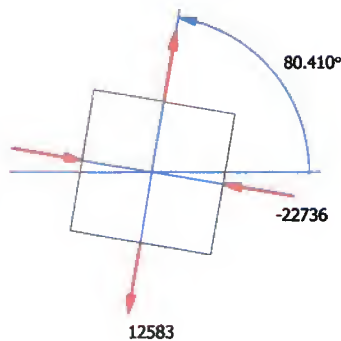
CCW

Angle of maximum shear stress

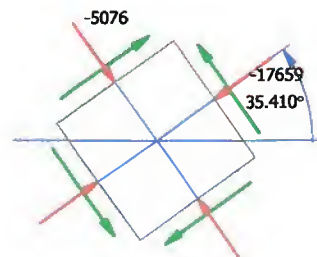
$$\phi_{\tau} = 35.410^\circ$$

CCW to $-\tau_{\max}$ 

Original stress element



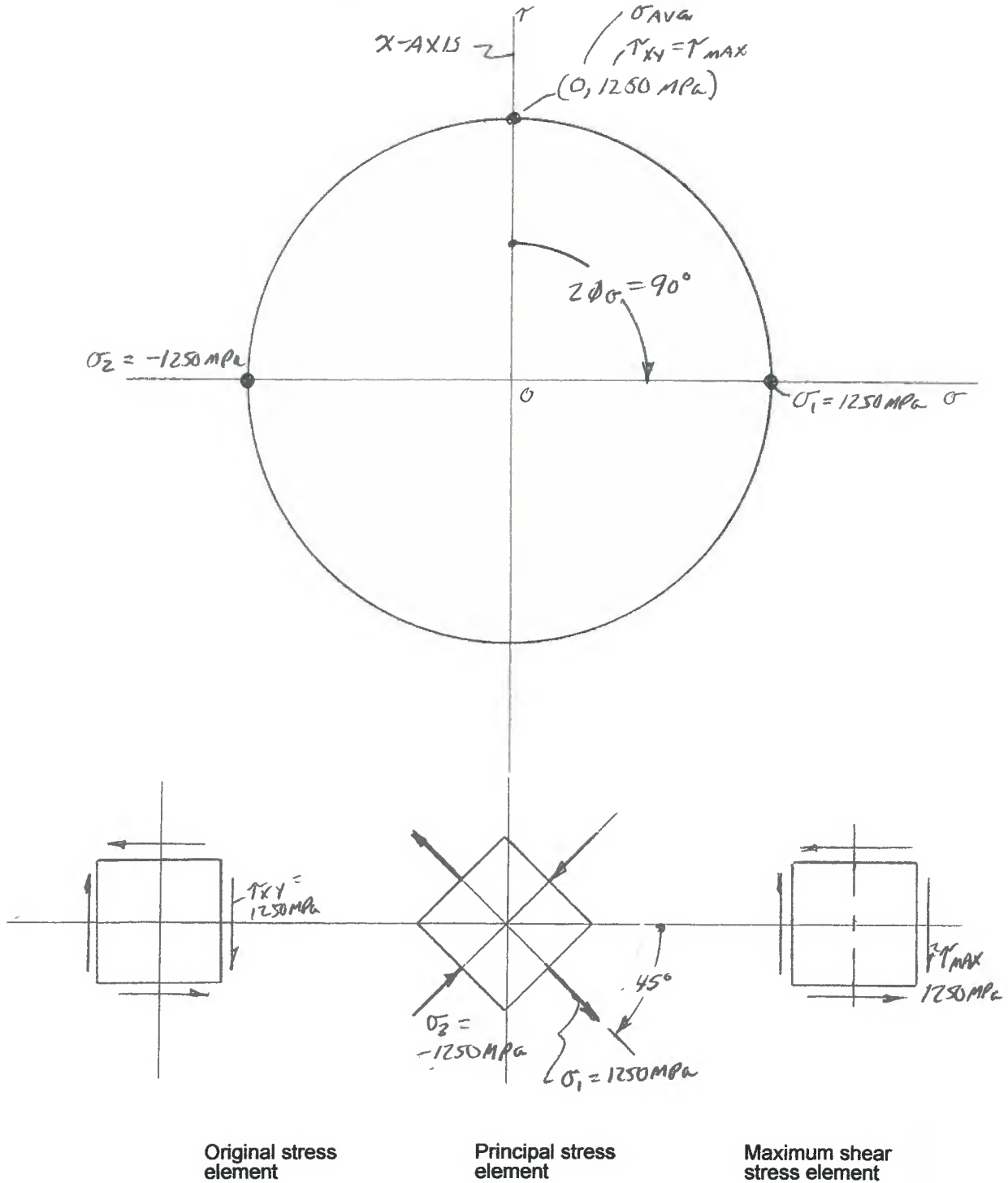
Principal stress element



Maximum shear stress element

14.

$$\sigma_x = 0, \sigma_y = 0, \tau_{xy} = 1250 \text{ MPa}$$

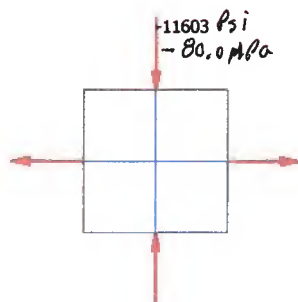
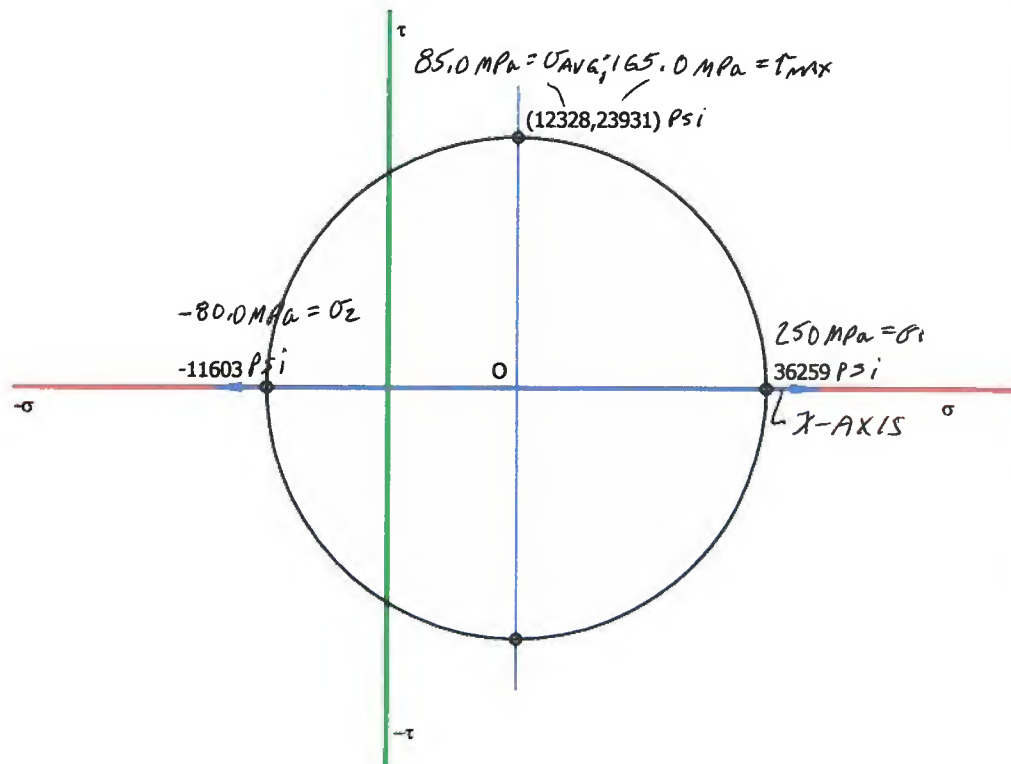


15

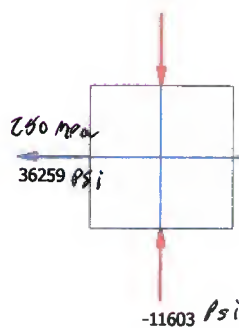
$\sigma_x = 250$ MPa
 $\sigma_y = -80$ MPa
 $\tau_{xy} = 0$ MPa

Results:

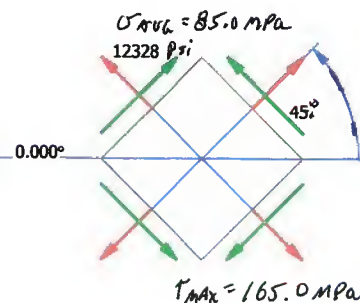
Maximum principal stress	$\sigma_1 =$	250.000	MPa
Minimum principal stress	$\sigma_2 =$	-80.000	MPa
Maximum shear stress	$\tau_{max} =$	165.000	MPa
Average normal stress	$\sigma_{avg} =$	85.000	MPa
Principal planes	$\phi_\sigma =$	0.000	°
			ccw
Angle of maximum shear stress	$\phi_\tau =$	45.000	°
			ccw to $-\tau_{max}$



Original stress element



Principal stress element



Maximum shear stress element

16

$\sigma_x = 50$ MPa
 $\sigma_y = -80$ MPa
 $\tau_{xy} = -30$ MPa

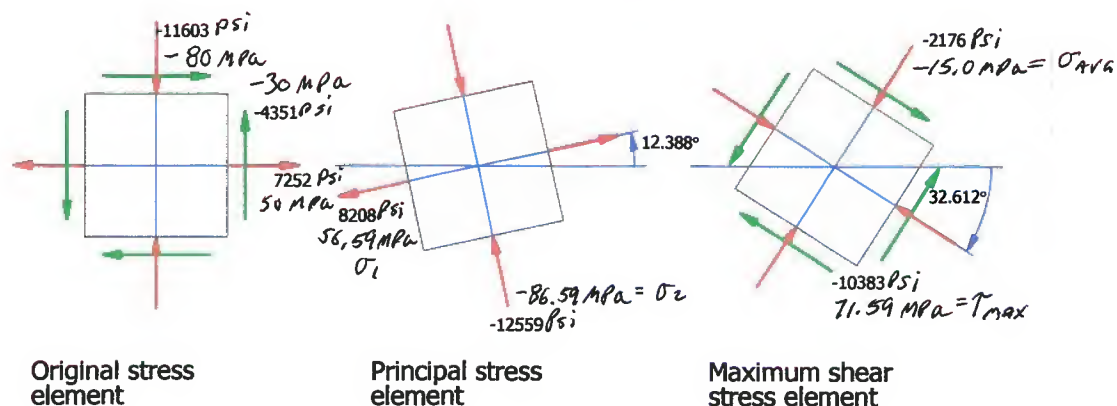
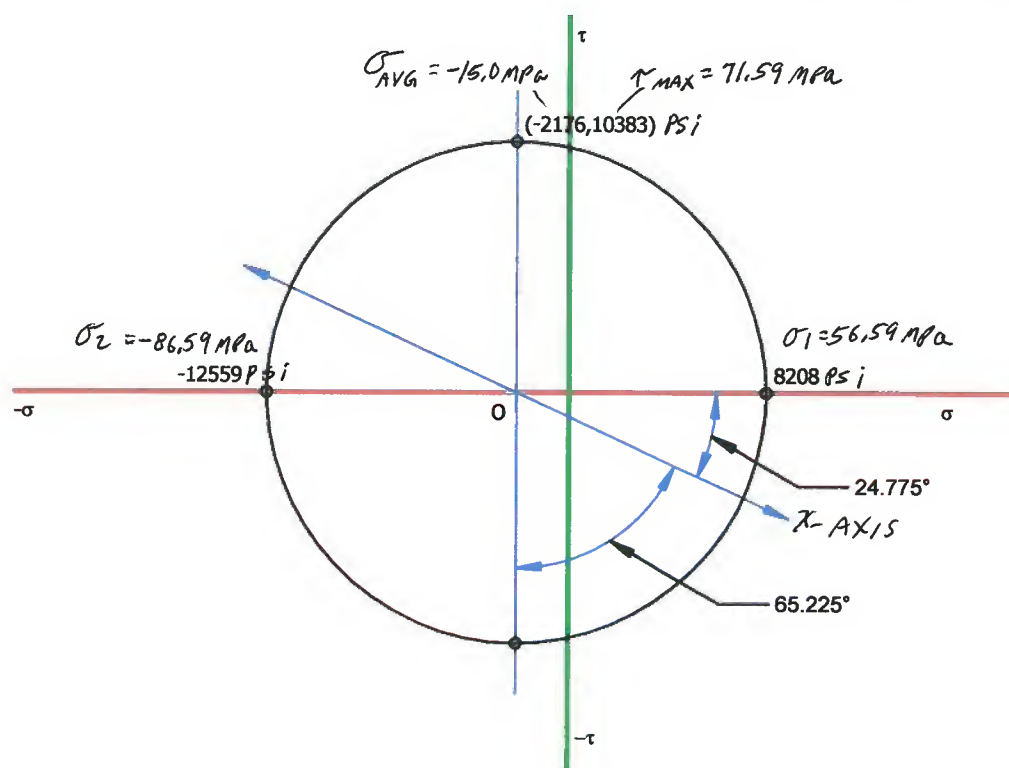
Results:

Maximum principal stress $\sigma_1 = 56.589$ MPa
 Minimum principal stress $\sigma_2 = -86.589$ MPa
 Maximum shear stress $\tau_{max} = 71.589$ MPa
 Average normal stress $\sigma_{avg} = -15.000$ MPa
 Principal planes $\phi_\sigma = 12.388^\circ$

Angle of maximum shear stress

 $\phi_\tau = 32.612^\circ$

CCW

CW to $-\tau_{max}$ 

17

$\sigma_x = 400$ MPa
 $\sigma_y = -300$ MPa
 $\tau_{xy} = 200$ MPa

Results:

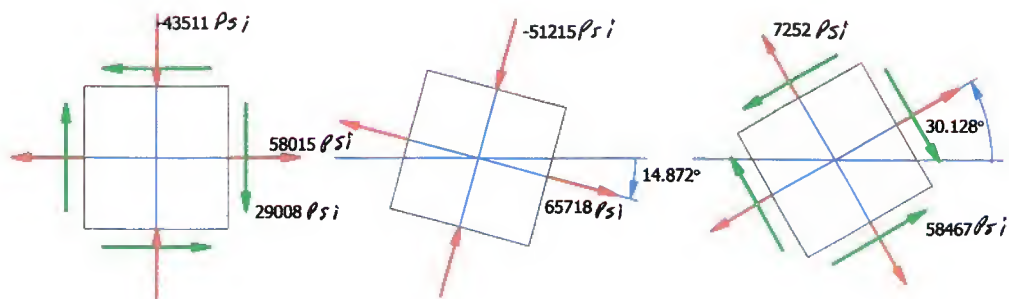
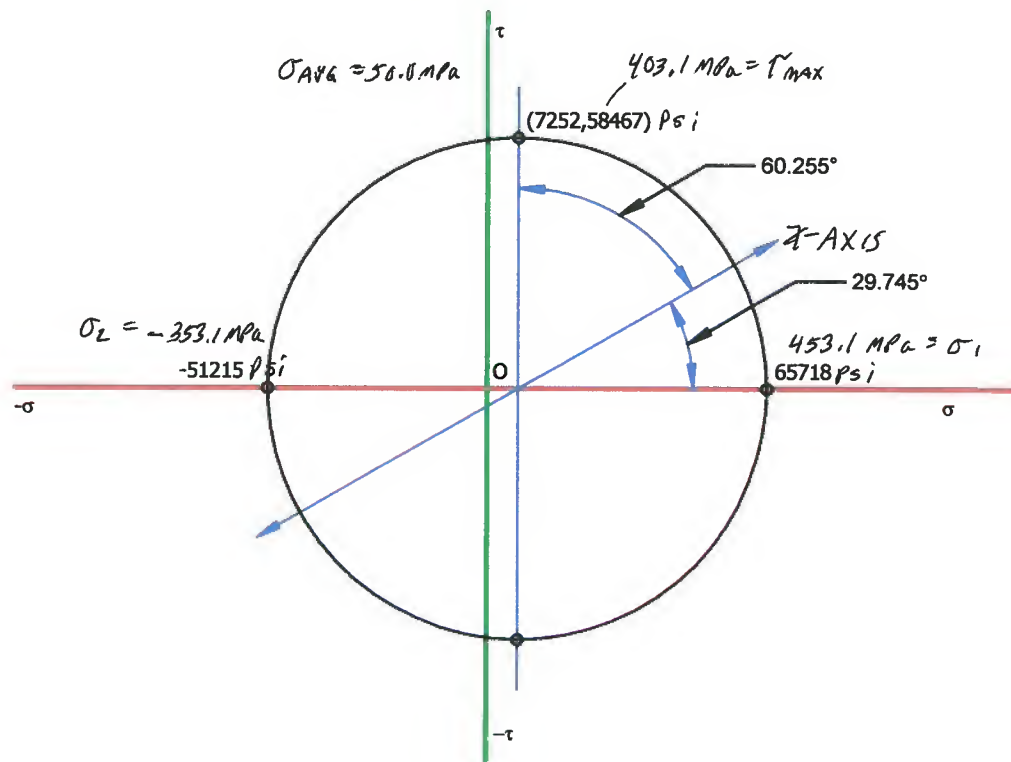
Maximum principal stress $\sigma_1 = 453.113$ MPa
 Minimum principal stress $\sigma_2 = -353.113$ MPa
 Maximum shear stress $\tau_{max} = 403.113$ MPa
 Average normal stress $\sigma_{avg} = 50.000$ MPa
 Principal planes $\phi_\sigma = 14.872^\circ$

Angle of maximum shear stress

$\phi_\tau = 30.128^\circ$

CW

CCW



Original stress element

Principal stress element

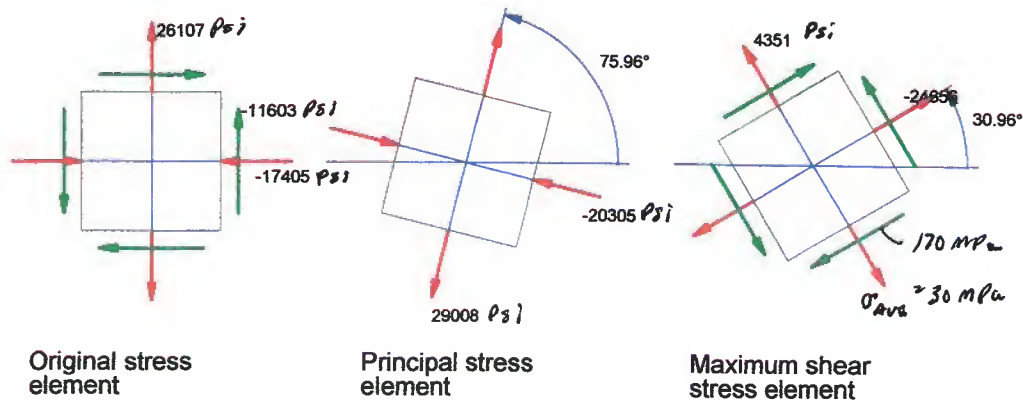
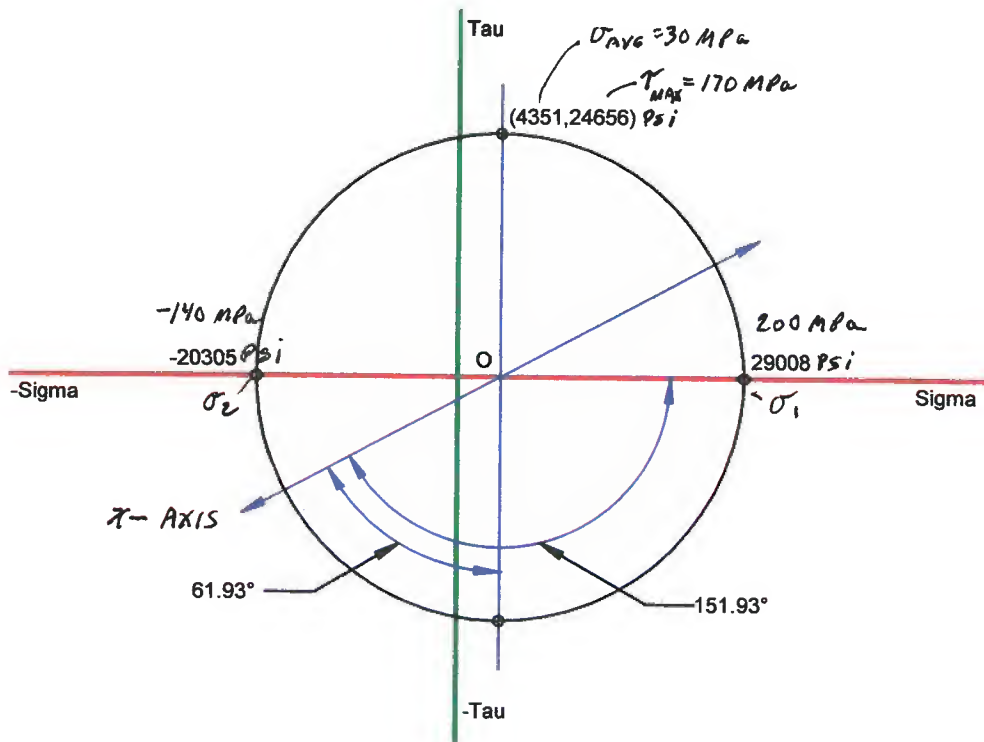
Maximum shear stress element

18.

$\sigma_x = -120$ MPa
 $\sigma_y = 180$ MPa
 $\tau_{xy} = -80$ MPa

Results

Maximum principal stress	σ_1	=	200.000 MPa
Minimum principal stress	σ_2	=	-140.000 MPa
Maximum shear stress	τ_{max}	=	170.000 MPa
Average normal stress	σ_{avg}	=	30.000 MPa
Principal planes	ϕ_σ	=	75.964°
			CCW
Angle of maximum shear stress	ϕ_τ	=	30.964°
			CCW to $-\tau_{max}$

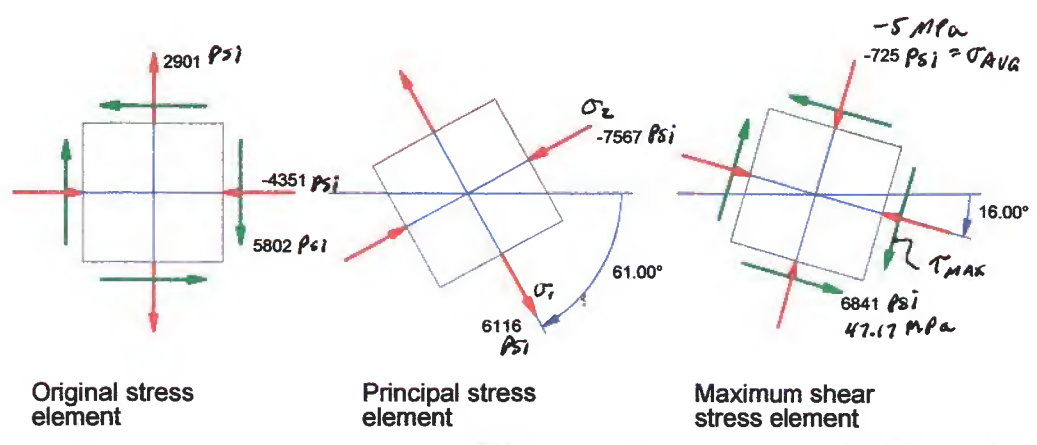
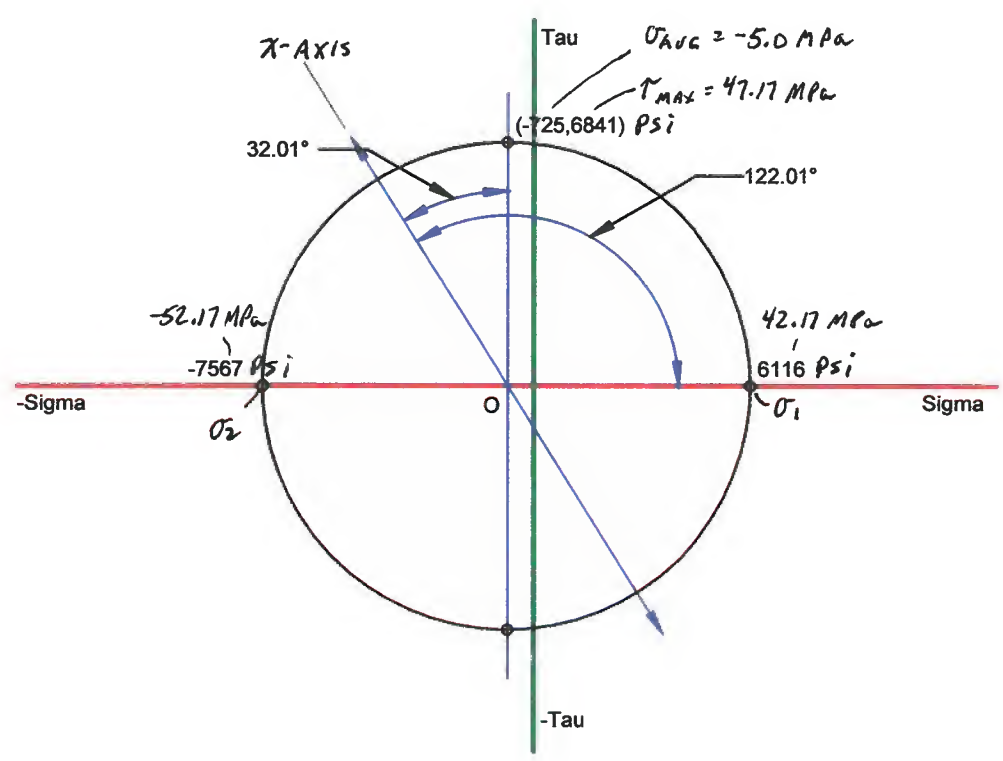


19,

$\sigma_x = -30$ MPa
 $\sigma_y = 20$ MPa
 $\tau_{xy} = 40$ MPa

Results

Maximum principal stress	σ_1	=	42.170 MPa
Minimum principal stress	σ_2	=	-52.170 MPa
Maximum shear stress	τ_{max}	=	47.170 MPa
Average normal stress	σ_{avg}	=	-5.000 MPa
Principal planes	ϕ_σ	=	61.003°
			CW
Angle of maximum shear stress	ϕ_τ	=	16.003°
			CW

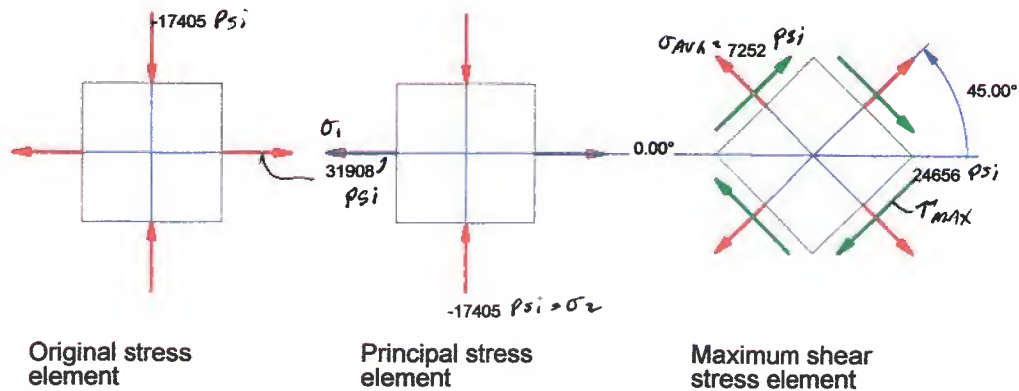
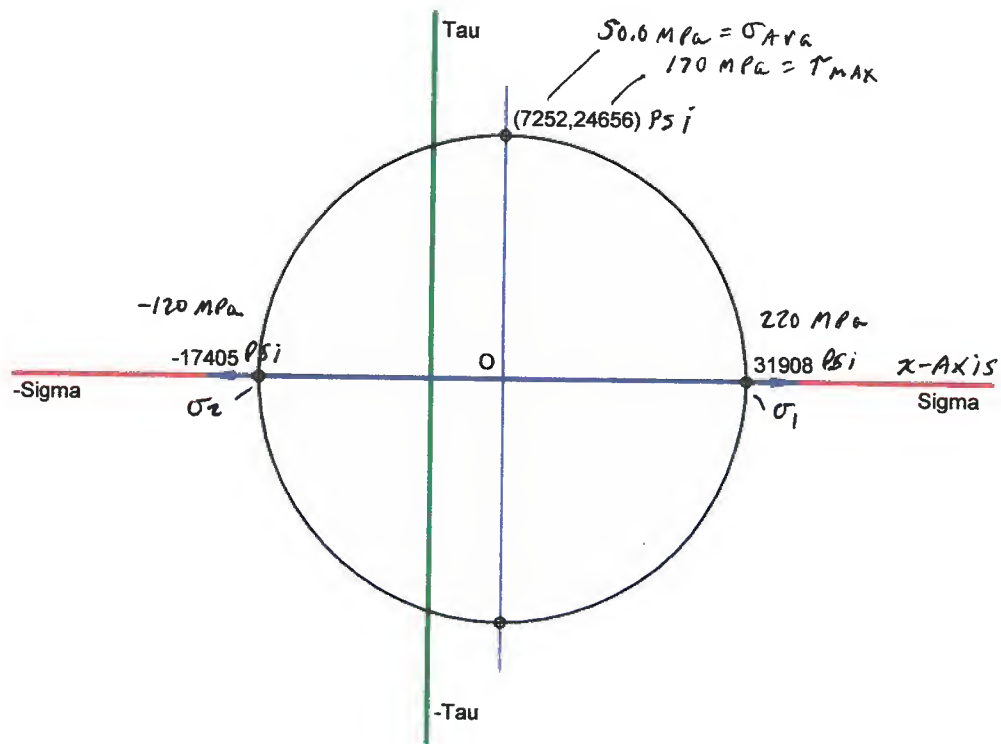


20.

$\sigma_x = 220$ MPa
 $\sigma_y = -120$ MPa
 $\tau_{xy} = 0$ MPa

Results

Maximum principal stress	σ_1	=	220.000 MPa
Minimum principal stress	σ_2	=	-120.000 MPa
Maximum shear stress	τ_{max}	=	170.000 MPa
Average normal stress	σ_{avg}	=	50.000 MPa
Principal planes	ϕ_σ	=	0.000°
			CCW
Angle of maximum shear stress	ϕ_τ	=	45.000°
			ccw to $-\tau_{max}$

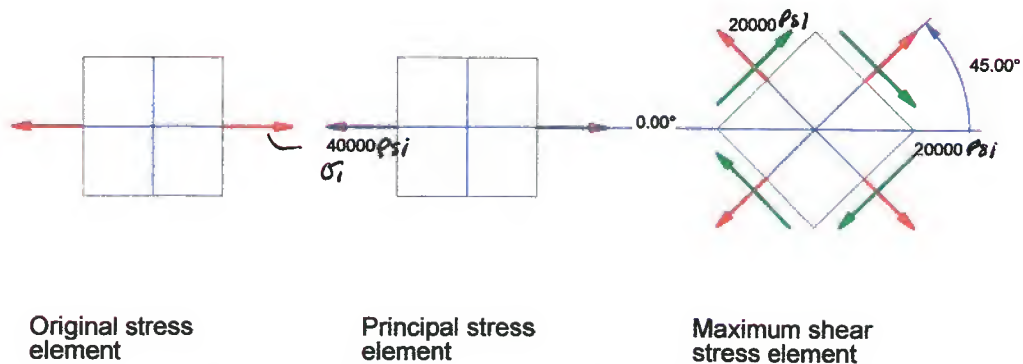
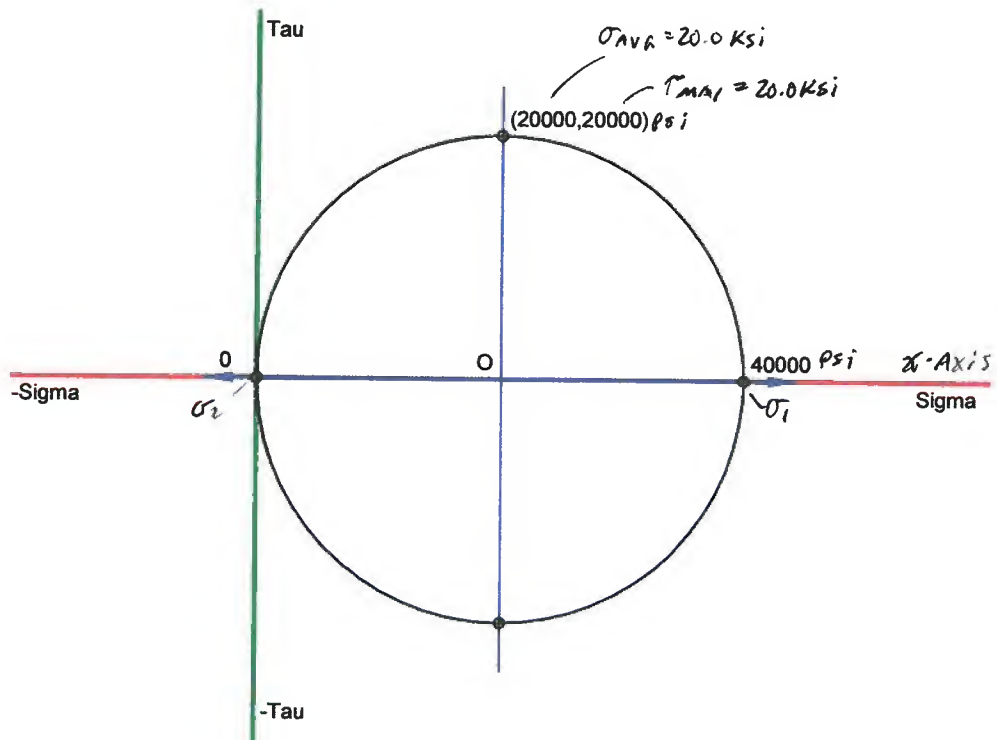


21,

$\sigma_x = 40$ Ksi
 $\sigma_y = 0$ Ksi
 $\tau_{xy} = 0$ Ksi

Results

Maximum principal stress	σ_1	=	40.000 Ksi
Minimum principal stress	σ_2	=	0.000 Ksi
Maximum shear stress	τ_{max}	=	20.000 Ksi
Average normal stress	σ_{avg}	=	20.000 Ksi
Principal planes	ϕ_σ	=	0.000°
			CCW
Angle of maximum shear stress	ϕ_τ	=	45.000°
			ccw to $-\tau_{max}$

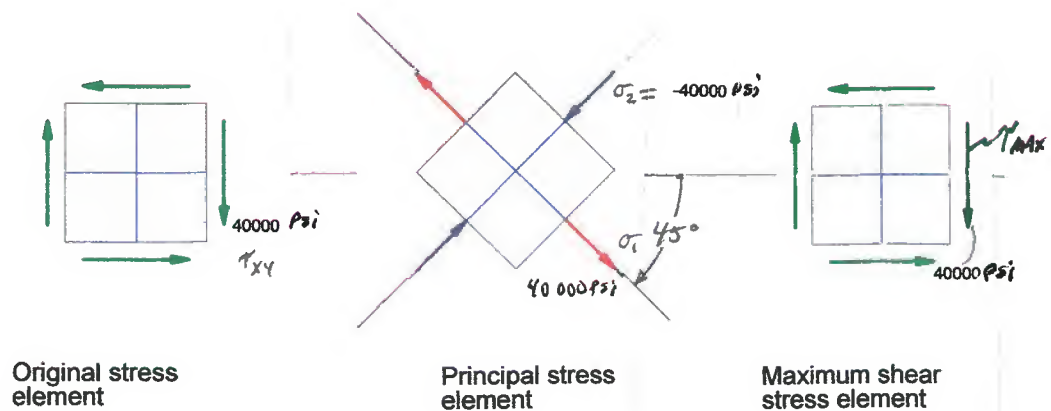
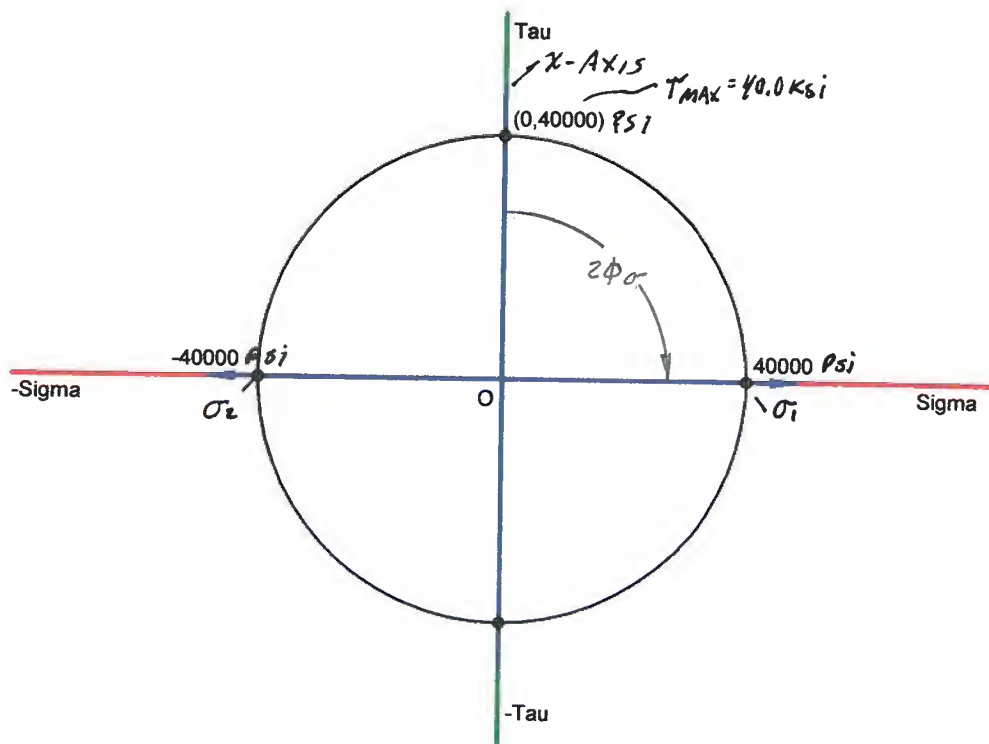


22.

$$\begin{aligned}\sigma_x &= 0 && \text{Ksi} \\ \sigma_y &= 0 && \text{Ksi} \\ \tau_{xy} &= 40 && \text{Ksi}\end{aligned}$$

Results

Maximum principal stress	σ_1	=	40.000 Ksi
Minimum principal stress	σ_2	=	-40.000 Ksi
Maximum shear stress	τ_{max}	=	40.000 Ksi
Average normal stress	σ_{avg}	=	0.000 psi
Principal planes	ϕ_σ	=	45.000°
Angle of maximum shear stress	ϕ_τ	=	0.000° CW CCW



23

$$\sigma_x = 38 \quad \text{ksi}$$

$$\sigma_y = -25 \quad \text{ksi}$$

$$\tau_{xy} = -18 \quad \text{ksi}$$

Results:

Maximum principal stress

$$\sigma_1 = 42.780 \quad \text{ksi}$$

Minimum principal stress

$$\sigma_2 = -29.780 \quad \text{ksi}$$

Maximum shear stress

$$\tau_{\max} = 36.280 \quad \text{ksi}$$

Average normal stress

$$\sigma_{\text{avg}} = 6.500 \quad \text{ksi}$$

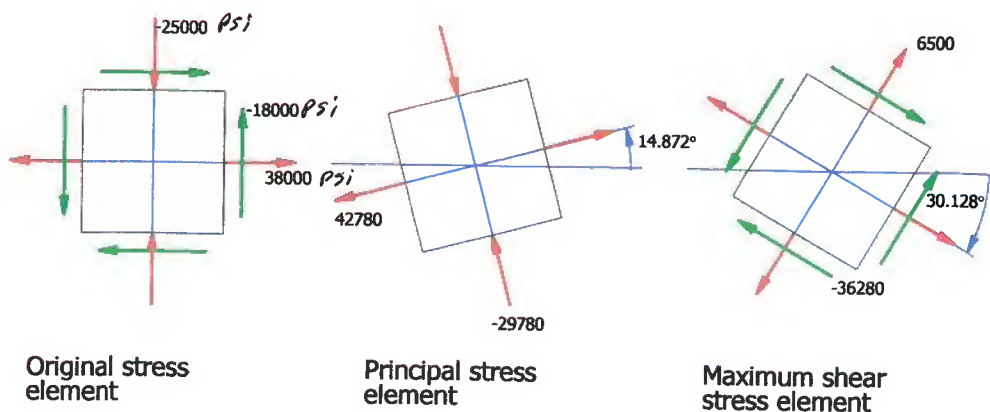
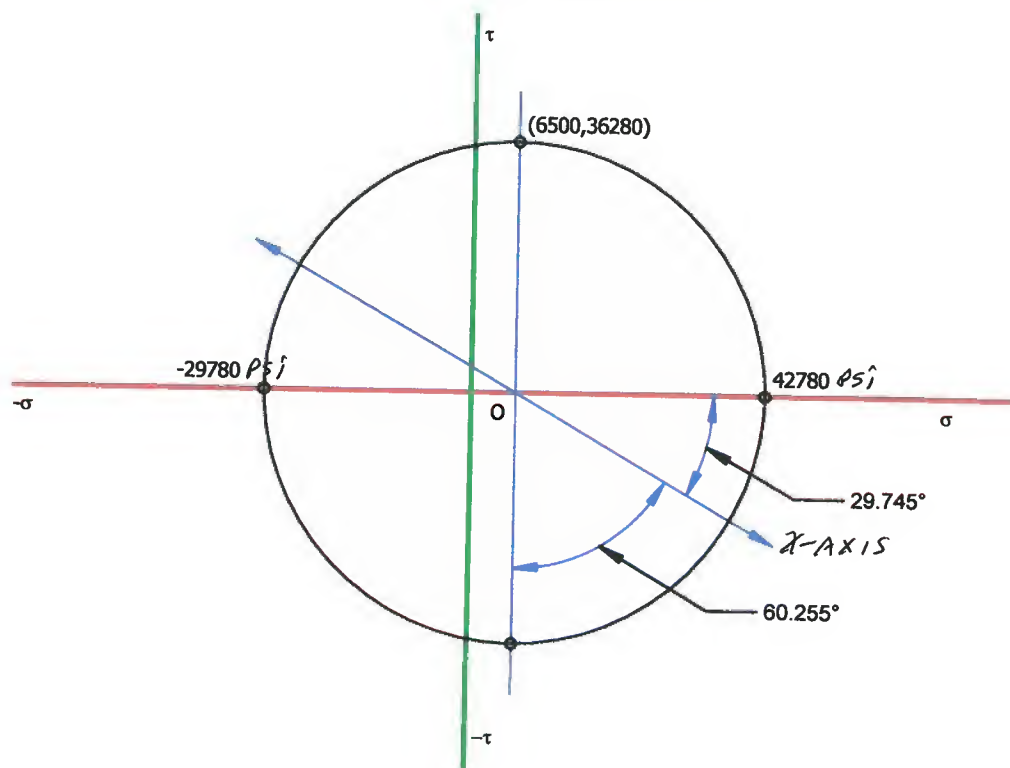
Principal planes

$$\phi_{\sigma} = 14.872 \quad ^\circ$$

CCW

Angle of maximum shear stress

$$\phi_{\tau} = 30.128 \quad ^\circ$$

CW to $-\tau_{\max}$ 

24

$$\sigma_x = 55 \quad \text{ksi}$$

$$\sigma_y = 0 \quad \text{ksi}$$

$$\tau_{xy} = 0 \quad \text{ksi}$$

Results:

Maximum principal stress

$$\sigma_1 = 55.000 \quad \text{ksi}$$

Minimum principal stress

$$\sigma_2 = 0.000 \quad \text{ksi}$$

Maximum shear stress

$$\tau_{\max} = 27.500 \quad \text{ksi}$$

Average normal stress

$$\sigma_{\text{avg}} = 27.500 \quad \text{ksi}$$

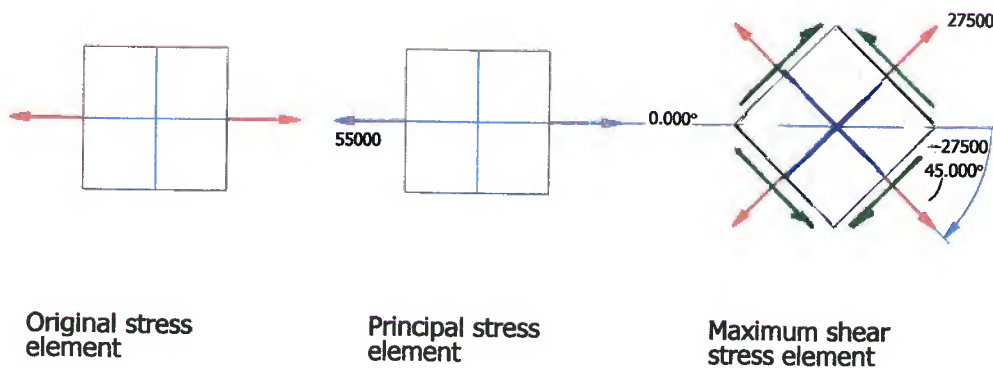
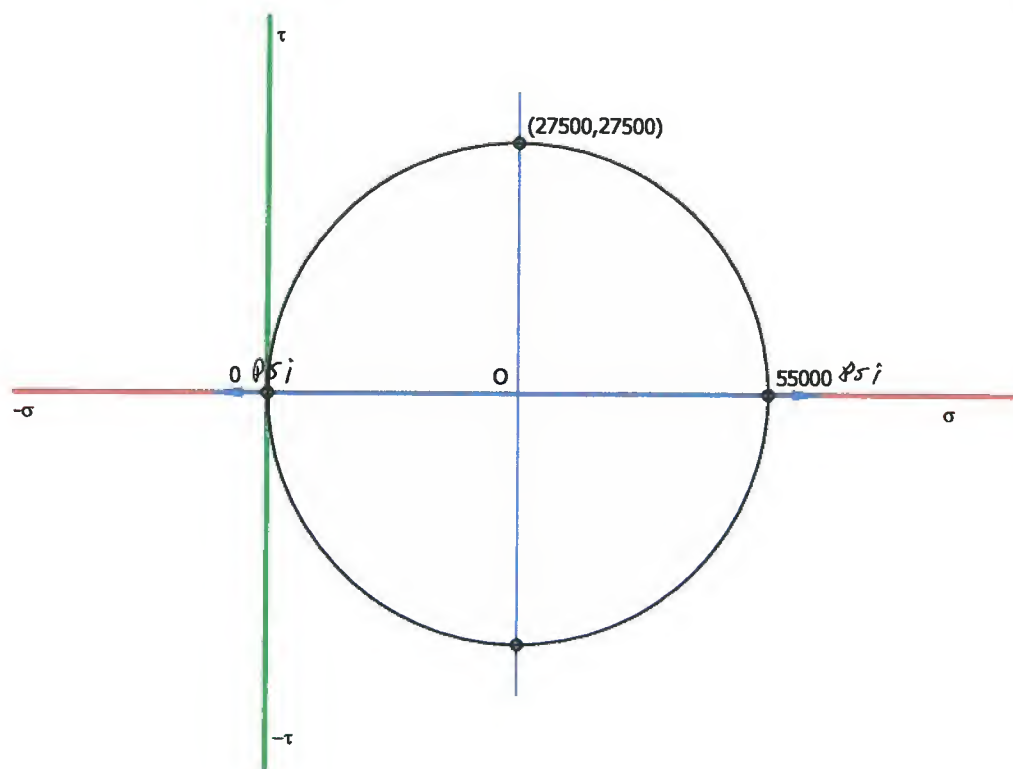
Principal planes

$$\phi_{\sigma} = 0.000 \quad ^\circ$$

Angle of maximum shear stress

$$\phi_{\tau} = 45.000 \quad ^\circ$$

CCW

CW to $-\tau_{\max}$ 

Original stress element

Principal stress element

Maximum shear stress element

25

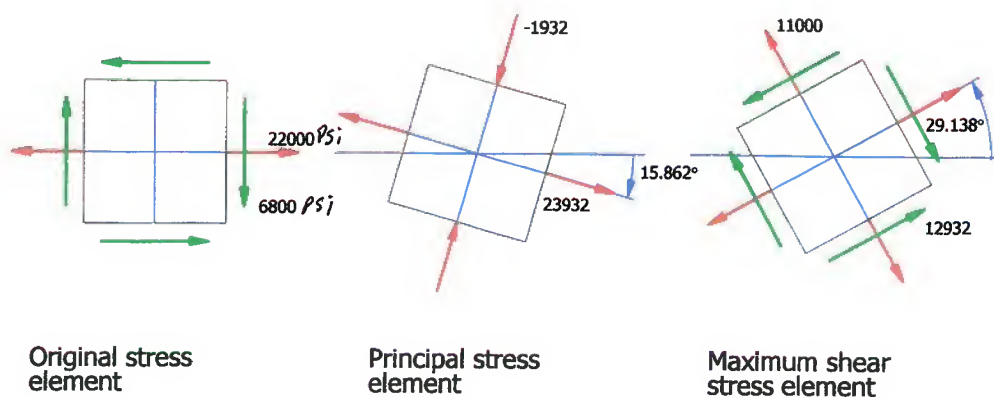
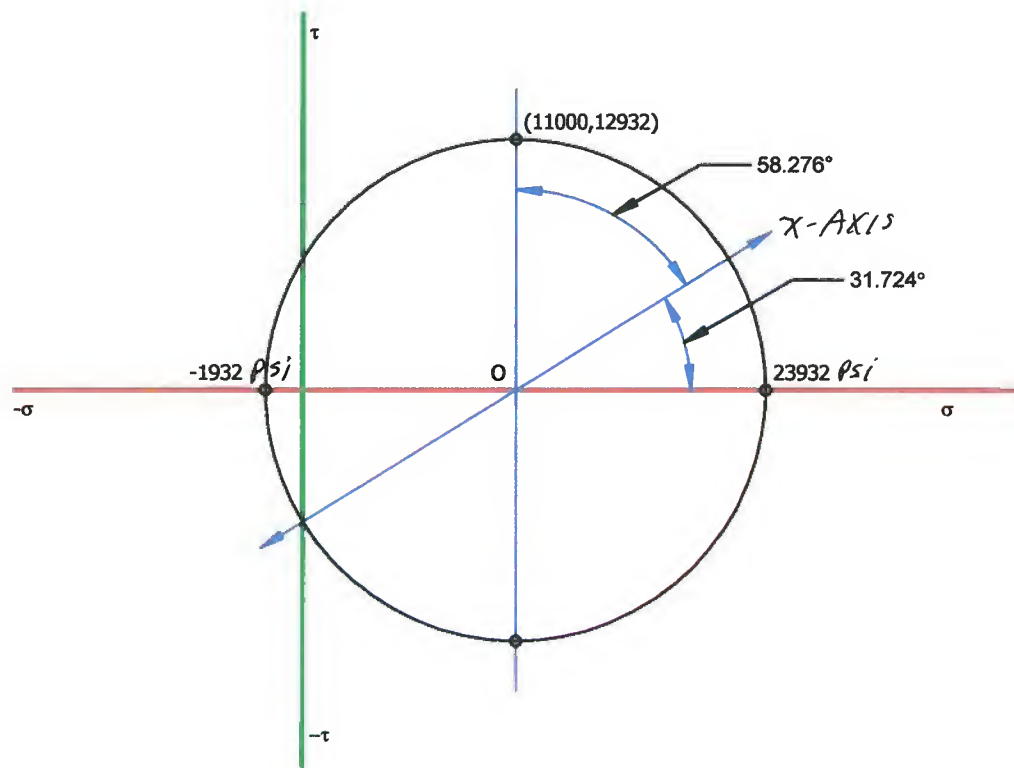
$\sigma_x = 22$ ksi
 $\sigma_y = 0$ ksi
 $\tau_{xy} = 6.8$ ksi

Results:

Maximum principal stress $\sigma_1 = 23.932$ ksi
 Minimum principal stress $\sigma_2 = -1.932$ ksi
 Maximum shear stress $\tau_{max} = 12.932$ ksi
 Average normal stress $\sigma_{avg} = 11.000$ ksi
 Principal planes $\phi_\sigma = 15.862^\circ$

Angle of maximum shear stress

$\phi_\tau = 29.138^\circ$
 CW
 CCW

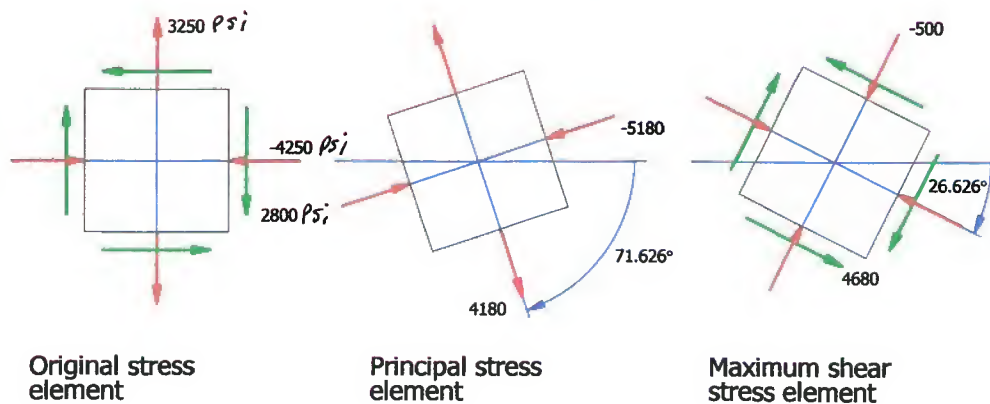
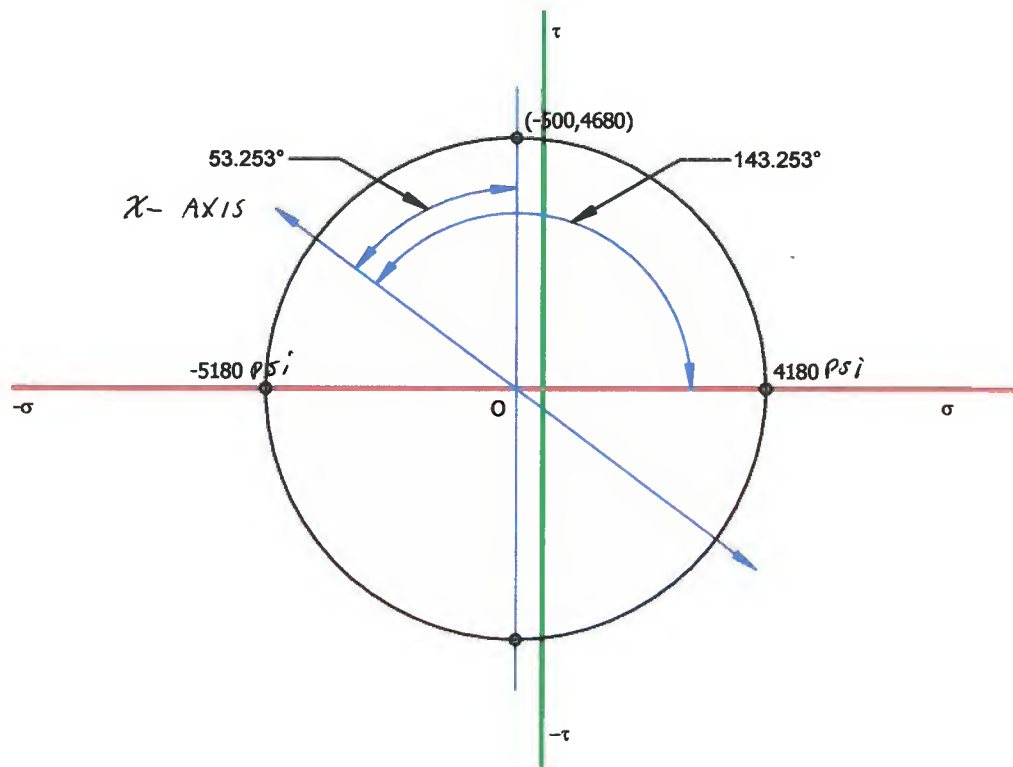


26

$$\begin{aligned}\sigma_x &= -4250 & \text{psi} \\ \sigma_y &= 3250 & \text{psi} \\ \tau_{xy} &= 2800 & \text{psi}\end{aligned}$$

Results:

Maximum principal stress	$\sigma_1 = 4180.011$	psi	
Minimum principal stress	$\sigma_2 = -5180.011$	psi	
Maximum shear stress	$\tau_{\max} = 4680.011$	psi	
Average normal stress	$\sigma_{\text{avg}} = -500.000$	psi	
Principal planes	$\phi_\sigma = 71.626$	°	
			CW
Angle of maximum shear stress	$\phi_\tau = 26.626$	°	
			CW



27

BOTH PRINCIPAL STRESSES ARE TENSILE - SAME SIGN

Input data:**Combined Stresses and Mohr's Circle**

Normal stress acting along x-axis
 Normal stress acting along y-axis
 Shear stress

$\sigma_x = 300$ MPa
 $\sigma_y = 100$ MPa
 $\tau_{xy} = 80$ MPa

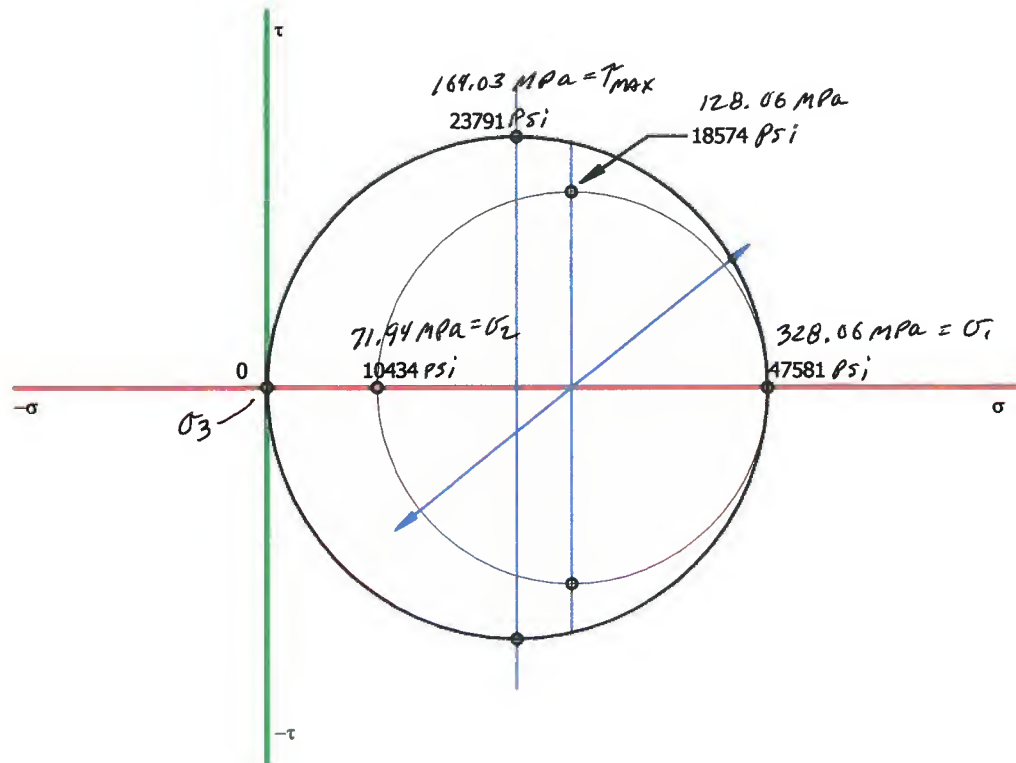
Results:

Maximum principal stress	σ_1	=	328.062	MPa
Minimum principal stress	σ_2	=	71.938	MPa
Minimum principal stress	σ_3	=	0.000	MPa
Maximum shear stress	τ_{max}	=	164.031	MPa
Shear stress	τ	=	128.062	MPa

Note: Both principle stresses have the same sign (both are tensile or both are compressive).

User must consider the resulting three-dimensional case.

Due to compound angles, elements as calculated are not applicable.



28

BOTH PRINCIPAL STRESSES ARE TENSILE - SAME SIGN.

Input data:**Combined Stresses and Mohr's Circle**

Normal stress acting along x-axis

 $\sigma_x = 250$

MPa

Normal stress acting along y-axis

 $\sigma_y = 150$

MPa

Shear stress

 $\tau_{xy} = 40$

MPa

Results:

Maximum principal stress

 $\sigma_1 = 264.031$

MPa

Minimum principal stress

 $\sigma_2 = 135.969$

MPa

Minimum principal stress

 $\sigma_3 = 0.000$

MPa

Maximum shear stress

 $\tau_{max} = 132.016$

MPa

Shear stress

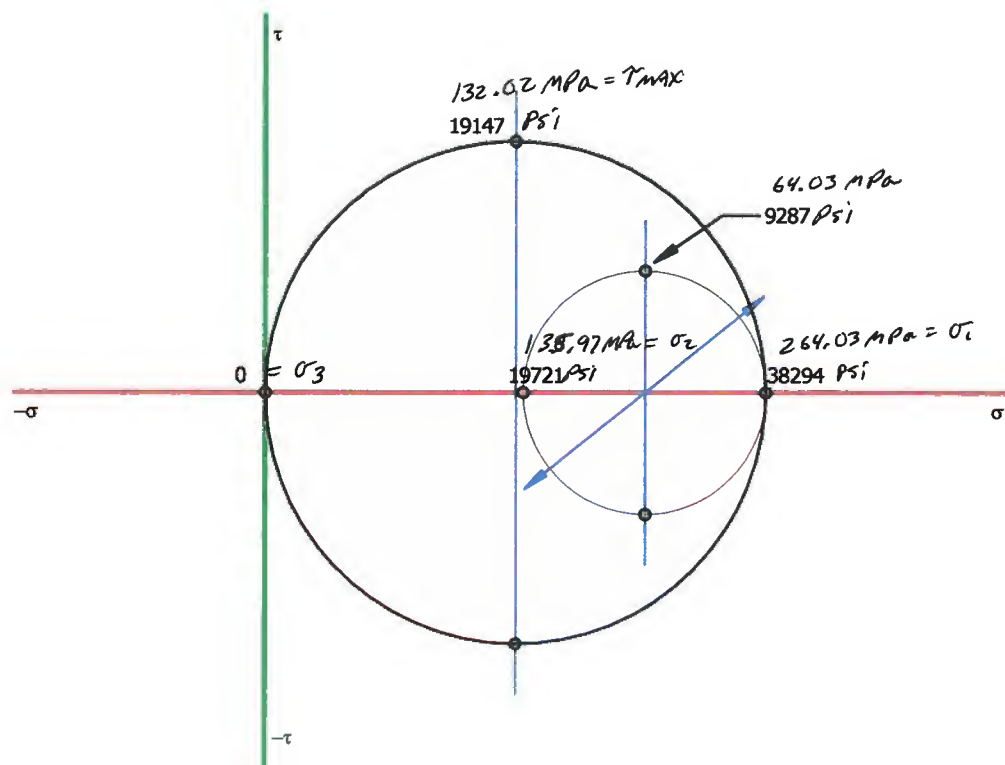
 $\tau = 64.031$

MPa

Note: Both principle stresses have the same sign (both are tensile or both are compressive).

User must consider the resulting three-dimensional case.

Due to compound angles, elements as calculated are not applicable.



29

BOTH PRINCIPAL STRESSES ARE COMPRESSIVE - SAME SIGN

Input data:**Combined Stresses and Mohr's Circle**

Normal stress acting along x-axis

 $\sigma_x = -840$ kPa

Normal stress acting along y-axis

 $\sigma_y = -335$ kPa

Shear stress

 $\tau_{xy} = -120$ kPa**Results:**

Maximum principal stress

 $\sigma_1 = 0.000$ kPa

Minimum principal stress

 $\sigma_2 = -307.936$ kPa

Minimum principal stress

 $\sigma_3 = -867.064$ kPa

Maximum shear stress

 $\tau_{max} = 433.532$ kPa

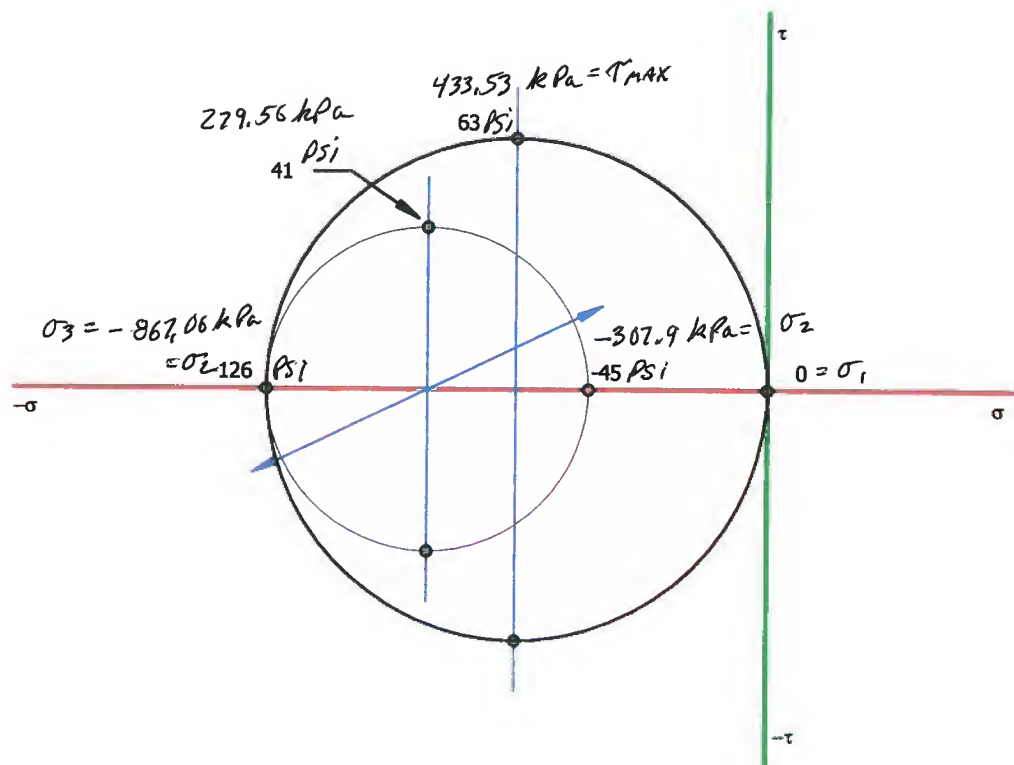
Shear stress

 $\tau = 279.564$ kPa

Note: Both principle stresses have the same sign (both are tensile or both are compressive).

User must consider the resulting three-dimensional case.

Due to compound angles, elements as calculated are not applicable.



30

BOTH PRINCIPAL STRESSES ARE COMPRESSIVE - SAME SIGN

Input data:**Combined Stresses and Mohr's Circle**

Normal stress acting along x-axis

 $\sigma_x = -325$ kPa

Normal stress acting along y-axis

 $\sigma_y = -50$ kPa

Shear stress

 $\tau_{xy} = -60$ kPa**Results:**

Maximum principal stress

 $\sigma_1 = 0.000$ kPa

Minimum principal stress

 $\sigma_2 = -37.479$ kPa

Minimum principal stress

 $\sigma_3 = -337.521$ kPa

Maximum shear stress

 $\tau_{max} = 168.760$ kPa

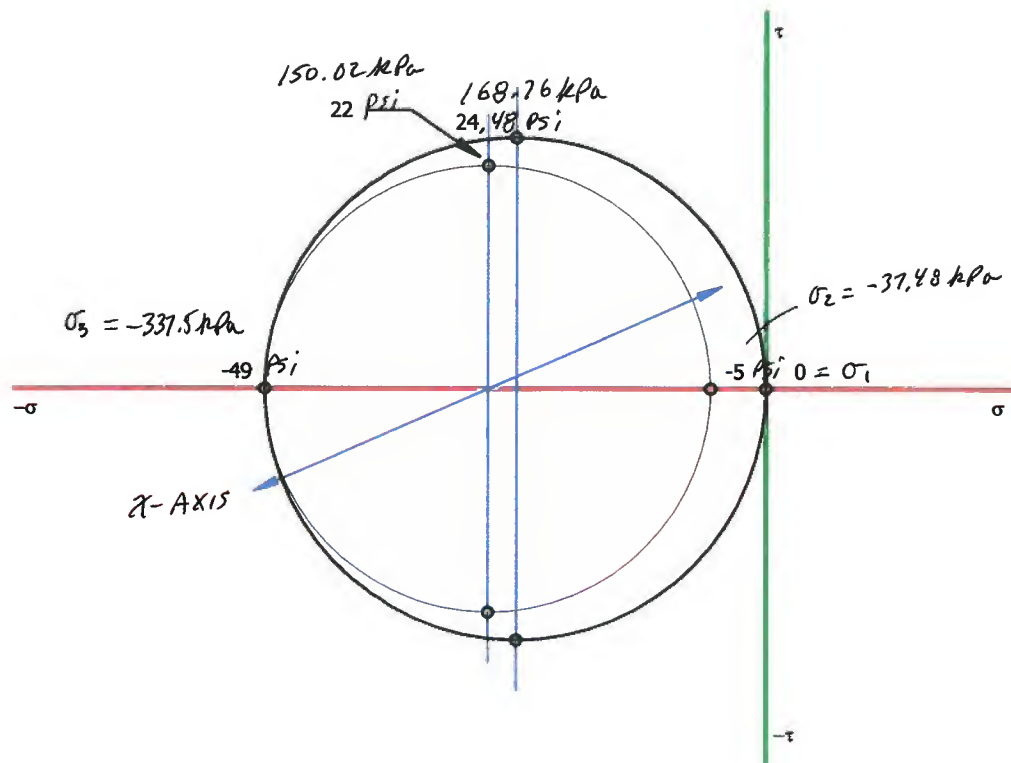
Shear stress

 $\tau = 150.021$ kPa

Note: Both principle stresses have the same sign (both are tensile or both are compressive).

User must consider the resulting three-dimensional case.

Due to compound angles, elements as calculated are not applicable.



31.

USE $D = 0.500$ IN FOR SHAFT ABC. STRESS ELEMENT ON BOTTOM.
FROM FIG. 3-23: $M_B = 252$ LB·IN.; $T_B = 300$ LB·IN.

$$Z = \pi D^3 / 32 = \pi (0.500 \text{ in})^3 / 32 = 0.01227 \text{ in}^3$$

$$\sigma_B = \frac{M_B}{Z} = \frac{252 \text{ LB·IN.}}{0.01227 \text{ in}^3} = 20538 \text{ psi} = \sigma_x$$

$$Z_p = \pi D^3 / 16 = 2Z = 0.02454 \text{ in}^3$$

$$\tau_B = \frac{T_B}{Z_p} = \frac{300 \text{ LB·IN.}}{0.02454 \text{ in}^3} = 12225 \text{ psi}$$

$$R = \tau_{MAX} = \sqrt{10269^2 + 12225^2} = 15966 \text{ psi}$$

$$\sigma_1 = 10269 + 15966 = 26235 \text{ psi}$$

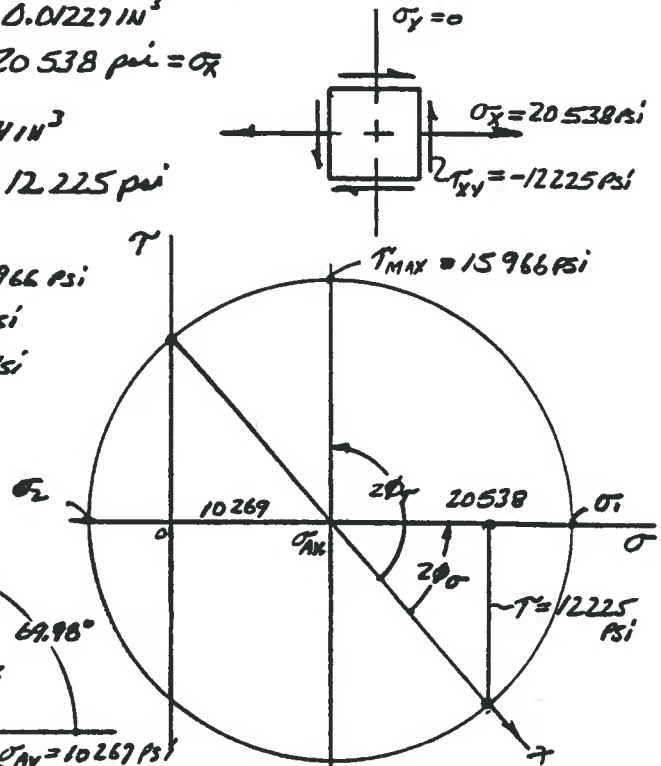
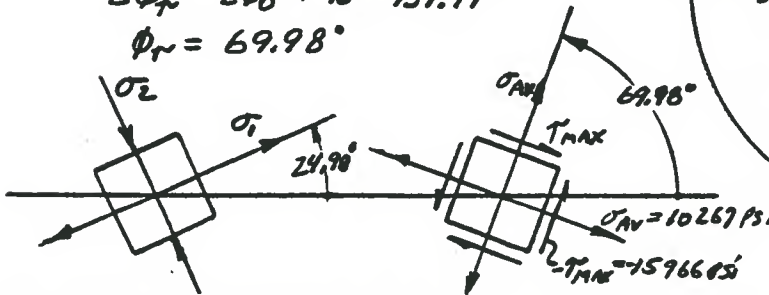
$$\sigma_2 = 10269 - 15966 = -5697 \text{ psi}$$

$$2\phi_\sigma = \tan^{-1} \left(\frac{12225}{10269} \right) = 49.97^\circ$$

$$\phi_\sigma = 24.98^\circ \text{ CCW}$$

$$2\phi_\tau = 2\phi_\sigma + 90^\circ = 139.97^\circ$$

$$\phi_\tau = 69.98^\circ$$



32.

USE $D = 1.500$ IN FOR SHAFT ABC. STRESS ELEMENT ON BOTTOM.

FROM SOLUTION FOR PROBLEM 3-48,

$$M_B = 4572 \text{ LB·IN.}; T_B = 6400 \text{ LB·IN.}$$

$$Z = \pi D^3 / 32 = 0.3313 \text{ in}^3; \sigma_B = M_B / Z = 4572 \text{ LB·IN.} / 0.3313 \text{ in}^3 = 13800 \text{ psi} = \sigma_x$$

$$Z_p = \pi D^3 / 16 = 2Z = 0.6627 \text{ in}^3; \tau_B = T_B / Z_p = 6400 / 0.6627 = 9658 \text{ psi}$$

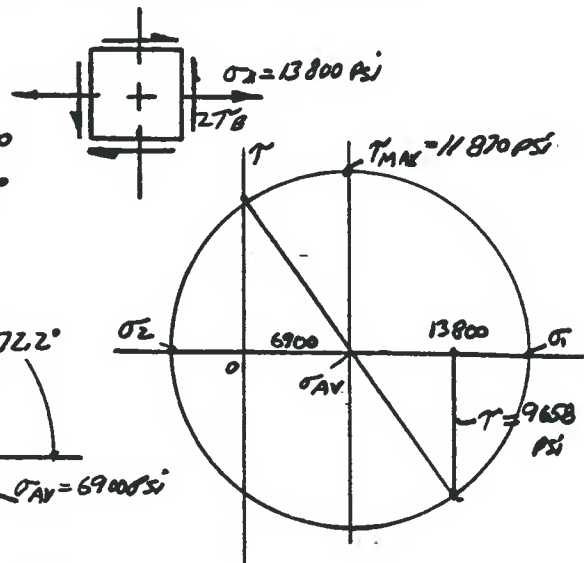
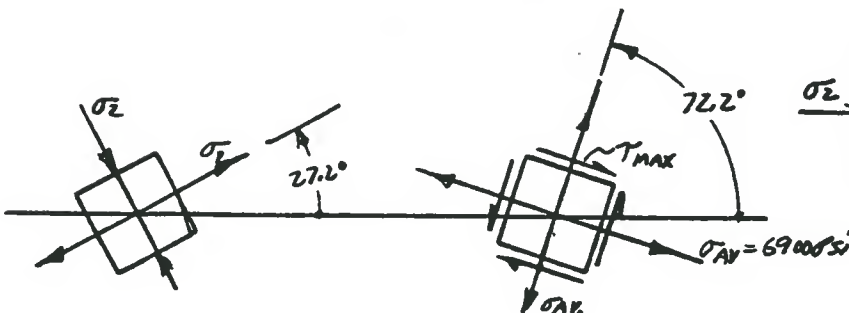
$$R = \tau_{MAX} = \sqrt{6900^2 + 9658^2} = 11870 \text{ psi}$$

$$\sigma_1 = 6900 + 11870 = 18770 \text{ psi}$$

$$\sigma_2 = 6900 - 11870 = -4970 \text{ psi}$$

$$2\phi_\sigma = \tan^{-1} \left(\frac{9658}{6900} \right) = 54.46^\circ; \phi_\sigma = 27.2^\circ$$

$$2\phi_\tau = 2\phi_\sigma + 90^\circ = 144.46^\circ; \phi_\tau = 72.2^\circ$$



33.

USE $D = 2.25 \text{ IN.}$ FOR SHAFT ABC.: STRESS ELEMENT ON BOTTOM.

FROM SOLUTION FOR PROBLEM 3-49,

$$M_B = 8640 \text{ LB}\cdot\text{IN.}; T_B = 400 \text{ LB}\cdot\text{IN.}$$

$$Z = \pi D^3/32 = 1.118 \text{ IN}^3; \sigma_B = M/Z = 8640 \text{ LB}\cdot\text{IN.}/1.118 \text{ IN}^3 = 7726 \text{ PSI TENSION}$$

$$Z_P = \pi D^3/16 = 2Z = 2.237 \text{ IN}^3; T_B = T_B/Z_P = 400 \text{ LB}\cdot\text{IN.}/2.237 \text{ IN}^3 = 179 \text{ PSI}$$

$$R = T_{\max} = \sqrt{179^2 + 3863^2} = 3867 \text{ psi}$$

$$\sigma_1 = 3863 + 3867 = 7730 \text{ psi}$$

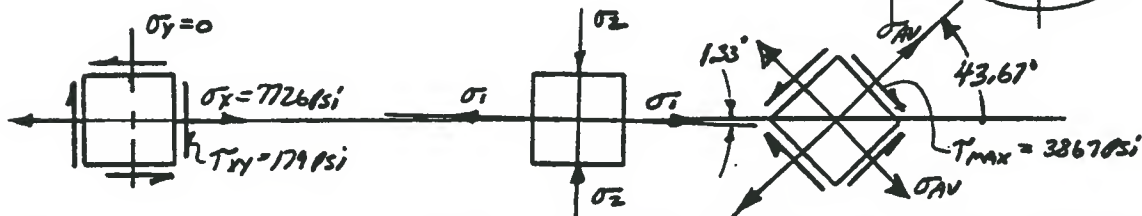
$$\sigma_2 = 3863 - 3867 = -4 \text{ psi}$$

$$2\phi_\sigma = \tan^{-1}(179/3863) = 2.65^\circ$$

$$\phi_\sigma = 1.33^\circ \text{ CCW}$$

$$2\phi_\tau = 90^\circ - 2\phi_\sigma = 87.35^\circ$$

$$\phi_\tau = 43.67^\circ \text{ CCW}$$



34.

USE $D = 50.0 \text{ mm}$ FOR SHAFT AB: STRESS ELEMENT ON BOTTOM.

FROM SOLUTION FOR PROBLEM 3-50,

$$M_A = -0.650 \text{ kN}\cdot\text{m}; T_A = 0.375 \text{ kN}\cdot\text{m}$$

$$Z = \pi D^3/32 = 12.272 \text{ mm}^3; \sigma_A = M/Z = \frac{650 \text{ N}\cdot\text{m}}{12.272 \text{ mm}^3} \cdot \frac{10^3 \text{ mm}^3}{\text{m}^3} = -52.91 \text{ MPa COMPRESSION}$$

$$Z_P = \pi D^3/16 = 2Z = 24.544 \text{ mm}^3; T_A = \frac{T}{Z_P} = \frac{375 \text{ N}\cdot\text{m}}{24.544 \text{ mm}^3} \cdot \frac{10^3 \text{ mm}^3}{\text{m}^3} = 15.28 \text{ MPa}$$

$$R = T_{\max} = \sqrt{15.28^2 + 26.48^2} = 30.57 \text{ MPa}$$

$$\sigma_1 = -26.48 + 30.57 = 4.09 \text{ MPa}$$

$$\sigma_2 = -26.48 - 30.57 = -57.05 \text{ MPa}$$

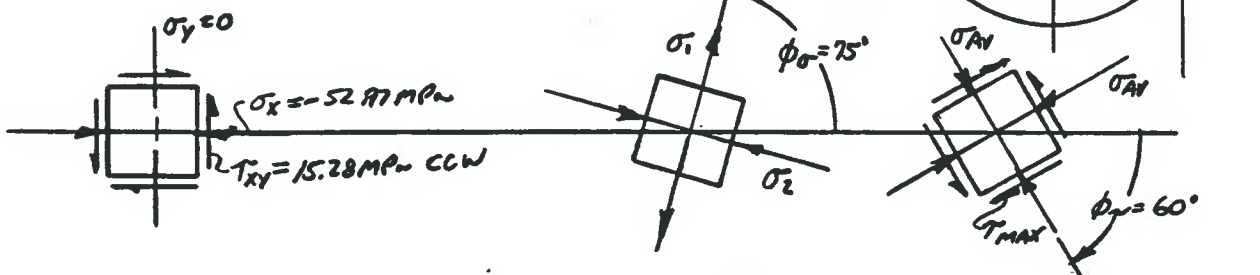
$$\alpha = \tan^{-1}(15.28/26.48) = 30.0^\circ$$

$$2\phi_\sigma = 180^\circ - \alpha = 150^\circ$$

$$\phi_\sigma = 75^\circ \text{ CCW}$$

$$2\phi_\tau = 90^\circ + \alpha = 120^\circ$$

$$\phi_\tau = 60^\circ \text{ CW}$$

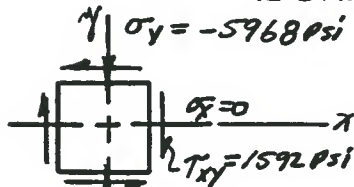


35.

$$D = 4.00 \text{ in} ; A = \pi D^2/4 = 12.57 \text{ in}^2 ; z_p = \pi D^3/16 = 12.57 \text{ in}^3$$

$$\sigma = \frac{-F}{A} = \frac{-75000 \text{ lb}}{12.57 \text{ in}^2} = -5968 \text{ psi}$$

$$\tau = \frac{T}{z_p} = \frac{20000 \text{ lb-in}}{12.57 \text{ in}^3} = 1592 \text{ psi}$$

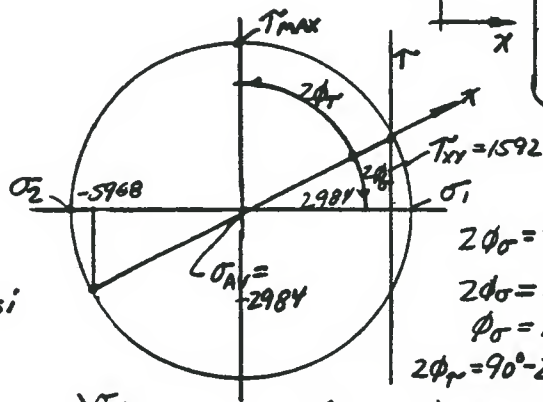
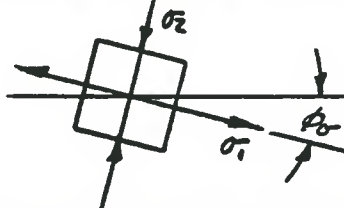


SURFACE ELEMENT

$$R = \tau_{MAX} = \sqrt{1592^2 + 2984^2} = 3382 \text{ psi}$$

$$\sigma_1 = -2984 + 3382 = 398 \text{ psi}$$

$$\sigma_2 = -2984 - 3382 = -6366 \text{ psi}$$



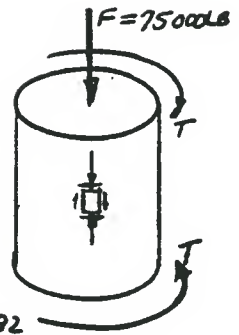
$$2\phi_\sigma = \tan^{-1}\left(\frac{1592}{2984}\right)$$

$$2\phi_\sigma = 28.08^\circ$$

$$\phi_\sigma = 14.04^\circ \text{ CW}$$

$$2\phi_\tau = 90^\circ - 2\phi_\sigma = 61.92^\circ$$

$$\phi_\tau = 30.96^\circ \text{ CCW}$$



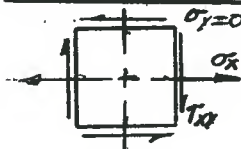
36.

$$D = 20 \text{ mm} ; A = \frac{\pi D^2}{4} = 314 \text{ mm}^2$$

$$\sigma_x = \frac{F}{A} = \frac{36000 \text{ N}}{314 \text{ mm}^2} = 114.6 \text{ MPa TENSION}$$

$$z_p = \frac{\pi D^3}{16} = 1571 \text{ mm}^3$$

$$\tau_{xy} = T/z_p = \frac{450 \text{ N-m}}{1571 \text{ mm}^3} \times \frac{10^3 \text{ mm}}{1 \text{ m}} = 286.48 \text{ MPa}$$



$$\sigma_{AVG} = \frac{114.6 + 0}{2} = 57.3 \text{ MPa}$$

$$R = \tau_{MAX} = \sqrt{57.3^2 + 286.5^2}$$

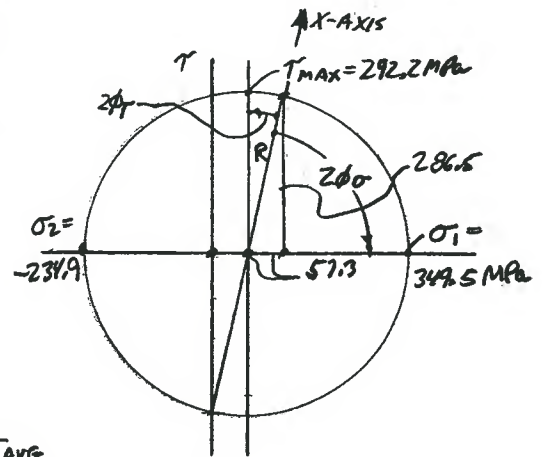
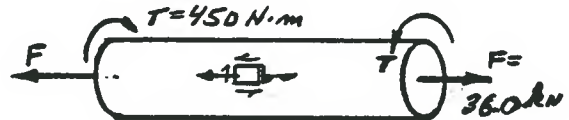
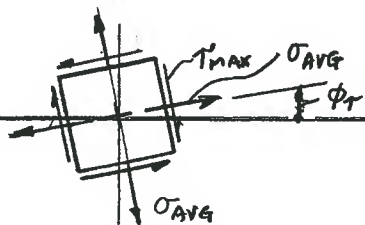
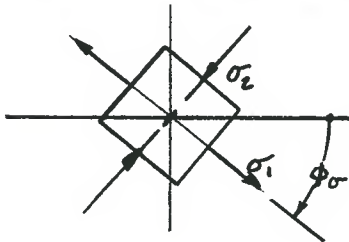
$$\tau_{MAX} = 292.2 \text{ MPa}$$

$$\sigma_1 = 57.3 + 292.2 = 349.5 \text{ MPa}$$

$$\sigma_2 = 57.3 - 292.2 = -234.9 \text{ MPa}$$

$$2\phi_\sigma = \tan^{-1}(286.5/57.3) = 78.7^\circ ; \phi_\sigma = 39.35^\circ$$

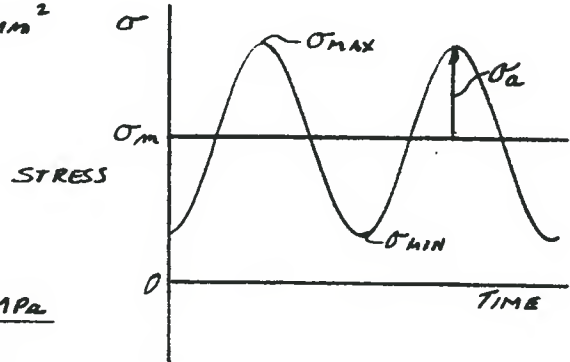
$$2\phi_\tau = 90^\circ - 2\phi_\sigma = 11.3^\circ ; \phi_\tau = 5.65^\circ$$



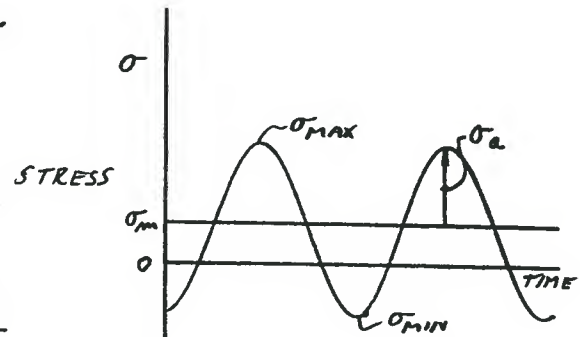
CHAPTER 5 DESIGN FOR DIFFERENT TYPES OF LOADING

Stress Ratio

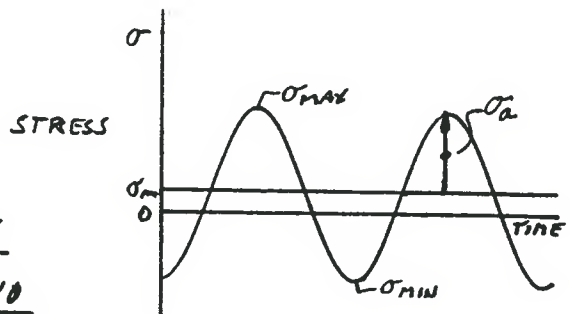
1. $\sigma = F/A$; $A = \frac{\pi(10\text{ mm})^2}{4} = 78.54\text{ mm}^2$
 $\sigma_{\text{MAX}} = 3500\text{ N} / 78.54\text{ mm}^2 = 44.6\text{ MPa}$
 $\sigma_{\text{MIN}} = 500\text{ N} / A = 6.37\text{ MPa}$
 $F_m = (3500 + 500) / 2 = 2000\text{ N}$
 $\sigma_m = 2000\text{ N} / A = 25.5\text{ MPa}$
 $\sigma_a = \sigma_{\text{MAX}} - \sigma_m = 44.6 - 25.5 = 19.1\text{ MPa}$
 $R = \sigma_{\text{MIN}} / \sigma_{\text{MAX}} = 6.37 / 44.6 = 0.143$



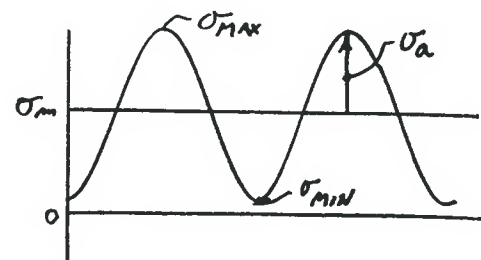
2. $\sigma = F/A$; $A = (10\text{ mm})(30\text{ mm}) = 300\text{ mm}^2$
 $\sigma_{\text{MAX}} = 20 \times 10^3\text{ N} / 300\text{ mm}^2 = 66.7\text{ MPa}$
 $\sigma_{\text{MIN}} = -8.0 \times 10^3\text{ N} / A = -26.7\text{ MPa}$
 $F_m = (20 - 8) / 2 = 6\text{ kN}$
 $\sigma_m = 6 \times 10^3\text{ N} / A = 20.0\text{ MPa}$
 $\sigma_a = \sigma_{\text{MAX}} - \sigma_m = 66.7 - 20.0 = 46.7\text{ MPa}$
 $R = \sigma_{\text{MIN}} / \sigma_{\text{MAX}} = -26.7 / 66.7 = -0.40$



3. $\sigma = F/A$; $A = (0.40\text{ in})^2 = 0.16\text{ in}^2$
 $\sigma_{\text{MAX}} = 860\text{ LB} / 0.16\text{ in}^2 = 5375\text{ psi}$
 $\sigma_{\text{MIN}} = -120\text{ LB} / 0.16\text{ in}^2 = -750\text{ psi}$
 $F_m = (860 - 120) / 2 = 370\text{ LB}$
 $\sigma_m = 370\text{ LB} / A = 2313\text{ psi}$
 $\sigma_a = \sigma_{\text{MAX}} - \sigma_m = 5375 - 2313 = 3062\text{ psi}$
 $R = \sigma_{\text{MIN}} / \sigma_{\text{MAX}} = -750 / 5375 = -0.140$



4. $\sigma = F/A$; $A = \pi D^2 / 4 = \pi (0.375)^2 / 4 = 0.1104\text{ in}^2$
 $\sigma_{\text{MAX}} = 1800\text{ LB} / 0.1104\text{ in}^2 = 16297\text{ psi}$
 $\sigma_{\text{MIN}} = 150\text{ LB} / 0.1104\text{ in}^2 = 1358\text{ psi}$
 $F_m = (1800 + 150) / 2 = 975\text{ LB}$
 $\sigma_m = 975\text{ LB} / A = 8828\text{ psi}$
 $\sigma_a = \sigma_{\text{MAX}} - \sigma_m = 16297 - 8828 = 7470\text{ psi}$
 $R = \sigma_{\text{MIN}} / \sigma_{\text{MAX}} = 1358 / 16297 = 0.083$



5. $\sigma = F/A$; $A = \frac{\pi d^2}{4} = \frac{\pi (3.0 \text{ mm})^2}{4} = 7.069 \text{ mm}^2$

$\sigma_{\text{MAX}} = 780 \text{ N} / 7.069 \text{ mm}^2 = 110.3 \text{ MPa}$

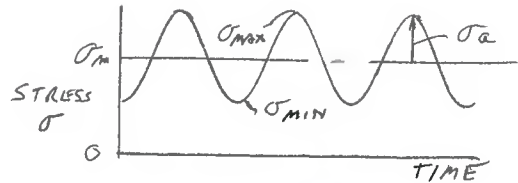
$\sigma_{\text{MIN}} = 360 \text{ N} / A = 50.9 \text{ MPa}$

$F_m = (180 + 360) / 2 = 570 \text{ N}$

$\sigma_m = 570 \text{ N} / A = 80.6 \text{ MPa}$

$\sigma_a = 110.3 - 80.6 = 29.7 \text{ MPa}$

$R = 50.9 / 110.3 = 0.462$



6.

$\sigma = M/S$; $S_y = 1.48 \text{ in}^3$ FOR $4 \times 2 \times 1/4$ TUBE σ_y APP 15-14 AT B

$\sigma_{\text{MAX}} = \frac{14400}{1.48} = 9730 \text{ psi}$

$\sigma_{\text{MIN}} = \frac{10560}{1.48} = 7135 \text{ psi}$

$\sigma_m = (9730 + 7135) / 2 = 8432 \text{ psi}$

$\sigma_a = \sigma_{\text{MAX}} - \sigma_m = 9730 - 8432 = 1298 \text{ psi}$

$\sigma_a = 2595 \text{ psi}$

$R = \sigma_{\text{MIN}} / \sigma_{\text{MAX}} = \frac{7135}{9730}$

$R = 0.733$

AT C
 $\sigma_{\text{MAX}} = \frac{14400}{1.48} = 9730 \text{ psi}$

$\sigma_{\text{MIN}} = \frac{3840}{1.48} = 2595 \text{ psi}$

$\sigma_m = (9730 + 2595) / 2 = 6162 \text{ psi}$

$\sigma_a = \sigma_{\text{MAX}} - \sigma_m = 9730 - 6162 = 3568 \text{ psi}$

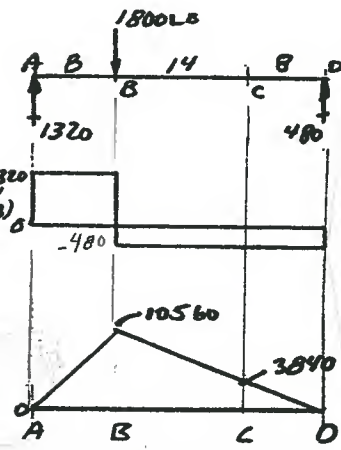
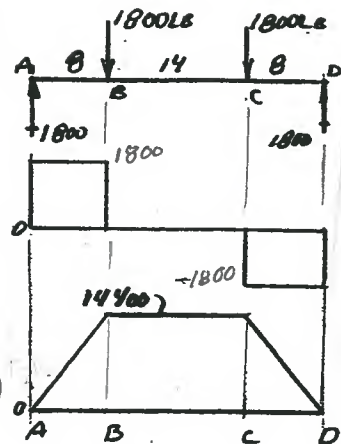
$\sigma_a = 3568 \text{ psi}$

$R = \sigma_{\text{MIN}} / \sigma_{\text{MAX}} = \frac{2595}{9730}$

$R = 0.267$

STRESS VS. TIME DIAGRAM - SAME AS FOR PROB. 5.

LOADING CASE I



7.

BEAM: $\sigma = \frac{M}{S}$; $S = 3.04 \text{ in}^3$ FOR 5×7.7

$M_{\text{MAX}} = (500 \text{ lb})(60 \text{ in}) = 30000 \text{ lb}\cdot\text{in} = M_1$

$M_{\text{MIN}} = (500 \text{ lb})(10 \text{ in}) = 5000 \text{ lb}\cdot\text{in} = M_2$

$M_m = (30000 + 5000) / 2 = 17500 \text{ lb}\cdot\text{in}$

$M_a = 30000 - 17500 = 12500 \text{ lb}\cdot\text{in}$

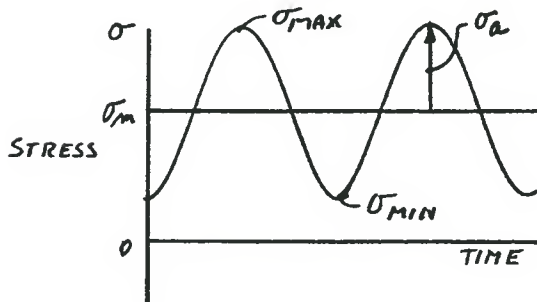
$\sigma_{\text{MAX}} = \frac{30000 \text{ lb}\cdot\text{in}}{3.04 \text{ in}^3} = 9868 \text{ psi}$

$\sigma_{\text{MIN}} = 5000 / 3.04 = 1645 \text{ psi}$

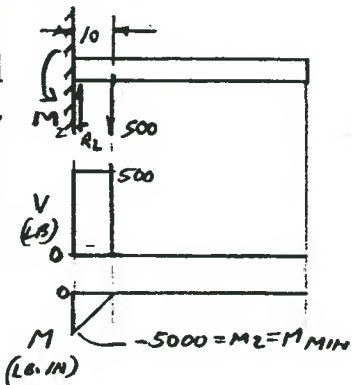
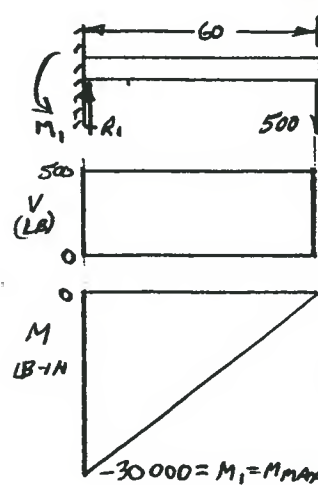
$\sigma_m = 17500 / 3.04 = 5757 \text{ psi}$

$\sigma_a = 12500 / 3.04 = 4112 \text{ psi}$

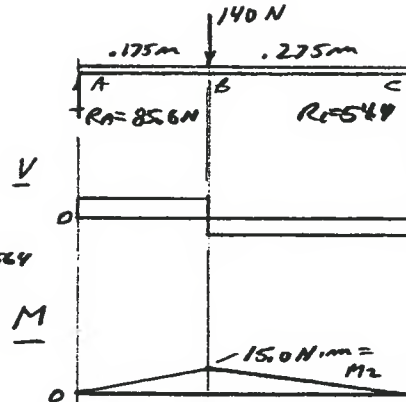
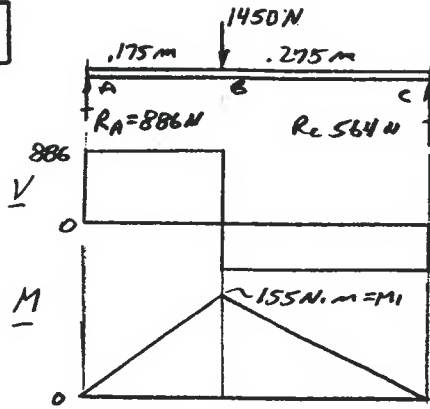
$R = \frac{\sigma_{\text{MIN}}}{\sigma_{\text{MAX}}} = \frac{1645}{9868} = 0.167$



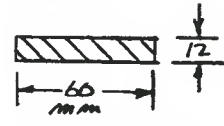
LOADING CASE II



8.



BEAM CROSS SECTION



$$S = \frac{BH^2}{6} = \frac{(60)(12)^2}{6}$$

$$S = 1440 \text{ mm}^3$$

$$\sigma = \frac{M}{S} \text{ BENDING}$$

$$\sigma_{\text{MAX}} = \frac{M_1}{S} = \frac{155 \text{ N}\cdot\text{m}}{1440 \text{ mm}^3} \cdot \frac{10^3 \text{ mm}^3}{\text{m}} = 107.7 \text{ MPa}$$

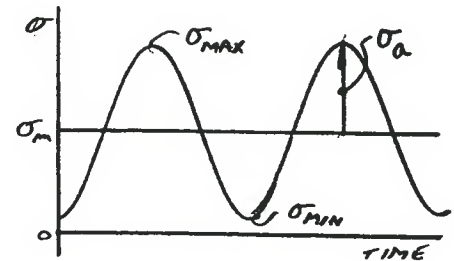
$$\sigma_{\text{MIN}} = \frac{M_2}{S} = \frac{15.0 \times 10^3 \text{ N}\cdot\text{mm}}{1440 \text{ mm}^3} = 10.4 \text{ MPa}$$

$$\sigma_m = (107.7 + 10.4)/2 = 59.1 \text{ MPa}$$

$$\sigma_a = \sigma_{\text{MAX}} - \sigma_m = 107.7 - 59.1 = 48.6 \text{ MPa}$$

$$R = \sigma_{\text{MIN}}/\sigma_{\text{MAX}} = 10.4/107.7 = 0.097$$

STRESS



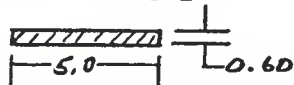
9.

SPRING IS A SUPPORTED CANTILEVER - CASE (b) APP A14-3.

DEFLECTION PROPORTIONAL TO FORCE. BENDING MOMENT PROPORTIONAL TO FORCE.

DEFLECTION AT LOAD B:

$$\eta_B = \frac{-Pa^3b^2}{12EIL^3}(3L+b)$$



$$I = \frac{(5)(.6)^3}{12} = 0.090 \text{ mm}^4$$

$$S = \frac{5(.6)^2}{6} = 0.30 \text{ mm}^3$$

$$E = 207 \times 10^3 \text{ N/mm}^2$$

SOLVE FOR P:

$$P = \frac{12EIL^3\eta_B}{a^3b^2(3L+b)} = \frac{(12)(207 \times 10^3)(0.090)(40)^3\eta_B}{(15)^3(25)^2[3(40)+25]} \text{ N}$$

$$P = 46.78 \text{ N}$$

FOR $\eta_1 = 0.25 \text{ mm}$; $P_1 = 46.78(0.25) = 11.7 \text{ N}$; FOR $\eta_2 = 0.40 \text{ mm}$, $P_2 = 18.7 \text{ N}$

MOMENTS:

$$M_A = \frac{-Pa^2b}{2L^2}(b+L) = \frac{-P(15)(25)}{2(40)^2}(25+40) = -7.617P \text{ MAXIMUM}$$

$$M_B = \frac{Pa^2b}{2L^3}(b+2L) = \frac{P(15)^2(25)}{2(40)^3}[25+2(40)] = 4.614P$$

SUMMARY

$$P_2 = 18.7 \text{ N}; M_{A2} = 7.617(18.7) = 142.5 \text{ N}\cdot\text{mm}; \sigma_{A2} = \frac{M}{S} = \frac{142.5 \text{ N}\cdot\text{mm}}{0.30 \text{ mm}^3} = 475 \text{ MPa} = \sigma_{\text{MAX}}$$

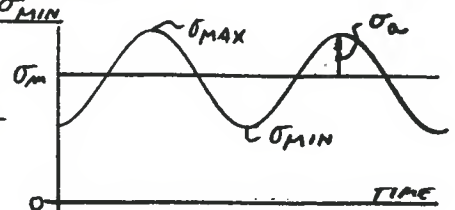
$$P_1 = 11.7 \text{ N}; M_{A1} = 89.1 \text{ N}\cdot\text{mm}; \sigma_{A1} = 297 \text{ MPa} = \sigma_{\text{MIN}}$$

$$\sigma_m = (475 + 297)/2 = 386 \text{ MPa}$$

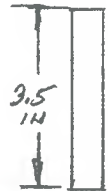
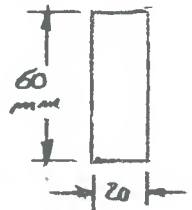
$$\sigma_a = 475 - 386 = 89.0 \text{ MPa}$$

$$R = \sigma_{\text{MIN}}/\sigma_{\text{MAX}} = 297/475 = 0.625$$

STRESS



10. FIND S_m' ; SAE 1040 CD; $S_u = 80 \text{ ksi}$; $S_m = 31 \text{ ksi}$ FOR CD CURVE
 0.75 IN DIA. $C_s = 0.90$ FIG. 5-9 FIG. 5-8.
 $C_m = 1.0$ WROUGHT STEEL; $C_{ST} = 1.0$ REV. BENDING; $C_R = 0.81$ ($R = 0.99$)
 $S_m' = S_m C_s C_m C_{ST} C_R = 31 \text{ ksi} (0.90)(1.0)(1.0)(0.81) = 22.6 \text{ ksi}$
11. FIND S_m' ; SAE 5160 OQT 1300; $S_u = 785 \text{ MPa}$; $S_m = 300 \text{ MPa}$ FIG. 5-8.
 20.0 mm DIA. $C_s = 0.90$ FIG. 5-9; $C_m = 1.0$ WROUGHT STEEL;
 $C_{ST} = 1.0$ REV. BENDING; $C_R = 0.81$ ($R = 0.99$)
 $S_m' = S_m C_s C_m C_{ST} C_R = 300 \text{ MPa} (0.90)(1.0)(1.0)(0.81) = 219 \text{ MPa}$
12. FIND S_m' ; SAE 4130 WQT 1300; $S_u = 676 \text{ MPa}$; $S_m = 260 \text{ MPa}$ FIG. 5-8
 EQ. (5-8); $D_e = 0.808 \sqrt{A b} = 0.808 \sqrt{(60)(20)} = 28.0 \text{ mm}$
 $C_s = 0.87$ FIG. 5-9; $C_m = 1.0$ WROUGHT STEEL
 $C_{ST} = 1.0$ REV. BENDING; $C_R = 0.81$ ($R = 0.99$)
 $S_m' = S_m C_s C_m C_{ST} C_R = 260 \text{ MPa} (0.87)(1.0)(1.0)(0.81) = 183 \text{ MPa}$
13. FIND S_m' ; SAE 301 ST. STL. $\frac{1}{2}$ HARD; $S_u = 150 \text{ ksi}$; $S_m = 52 \text{ ksi}$ FIG. 5-8.
 $C_s = 1.0$ FOR AXIAL TENSILE STRESS. $C_m = 1.0$ WROUGHT STEEL
 $C_{ST} = 0.80$ AXIAL TENSILE STRESS. $C_R = 0.75$ FOR $R = 0.999$.
 $S_m' = S_m C_s C_m C_{ST} C_R = 52 \text{ ksi} (1.0)(1.0)(0.80)(0.75) = 31.2 \text{ ksi}$
14. FIND S_m' ; ASTM A242; $S_u = 70 \text{ ksi}$; $S_m = 27.0 \text{ ksi}$, FIG. 5-8
 EQ. (5-8); $D_e = 0.808 \sqrt{A b} = 0.808 \sqrt{(3.5)(0.375)} = 0.926$; $C_s = 0.883$
 $C_m = 1.0$ WROUGHT STEEL; $C_{ST} = 1.0$ REV. BENDING; $C_R = 0.81$ ($R = 0.99$)
 $S_m' = S_m C_s C_m C_{ST} C_R = (27.0 \text{ ksi})(0.883)(1.0)(1.0)(0.81) = 19.3 \text{ ksi}$



Design and Analysis

Problems 15 - 18 are open-ended design problems for which there is no unique answer. The General Design Procedure from Section 5-9 should be used. The loading and support conditions should be compared with the cases described in Section 5-8 to determine the appropriate design stress. A design factor should be specified using the guidelines in Section 5-7. When needed, the endurance strength should be computed from Equation 5-6 in Section 5-4.

15. The link is subjected to a fluctuating normal stress. Use Case 5 from Section 5-8. See also the solution for Problem 1.
16. The rod is subjected to a fluctuating normal stress. Use Case 5 from Section 5-8. See also the solution for Problem 4.
17. The strut is subjected to a fluctuating normal stress. Use Case 5 from Section 5-8. See also the solution for Problem 2.
18. The latch part is subjected to a fluctuating normal stress. Use Case 5 from Section 5-8. See also the solution for Problem 5.

19.

FIG. P5-8. SEE ALSO PROBLEM 8 SOLUTION. FLUCTUATING LOAD

$$\frac{1}{N} = \frac{\sigma_m + K_t \sigma_a}{S_m} \quad \text{CASE 5} \quad \sigma_m = 59.1 \text{ MPa}; \sigma_a = 48.6 \text{ MPa}$$

FIND N . SAE 1020 HR; $S_y = 207 \text{ MPa}$; $S_m = 379 \text{ MPa}$; $S_u = 140 \text{ MPa}$ C_s: RECTANGLE: $D_e = 0.808 \sqrt{12(60)} = 21.7 \text{ mm}$; $C_s = 0.89$

FIG. 5-8 HOT ROL.

 $C_m = 1.0$ (WELDED HT STEEL), $C_{st} = 1.0$ (REV. BENDING); $LET R = 0.99 - C_R = 0.81$

$$S_m' = (0.89)(1.0)(1.0)(0.81)(140) = 101 \text{ MPa}$$

$$\text{YIELD: } N = \frac{S_y}{K_t(\sigma_m + \sigma_a)}$$

$$\frac{1}{N} = \frac{59.1}{379} + \frac{(1.0)(48.6)}{101} = 0.637; \quad N = 1.57$$

$$N_y = \frac{207}{1(59.1 + 48.6)} = 1.92$$

LOW

20.

DESIGN PROBLEM - NO UNIQUE SOLUTION.

SUGGESTIONS: KEEP 60 mm WIDTH FOR SEAT APPLICATION.

CONSIDER HIGHER STRENGTH MATERIAL; THINNER STOCK FORMED INTO CHANNEL SHAPE . CONSIDER TAPERING CROSS SECTION

DEPTH - DEEPER AT LOAD - LESS DEEP NEAR SUPPORTS WHERE MOMENT IS SMALLER. CONSIDER REF. R - APP. 15-7 OR REF. 9 - APP. 15-8. ALUMINUM.

21.

DATA ARE SAME AS IN PROB. 7 WHERE AN S4X7.7 STEEL BEAM WAS PROPOSED. FLUCTUATING STRESS. USE CASE 5. $A_T = 2.26 \text{ in}^2$

$$\frac{1}{N} = \frac{\sigma_m + K_t \sigma_a}{S_m} \quad \text{SPECIFY ASTM A36 STEEL } S_y = 36 \text{ ksi}; S_u = 58 \text{ ksi}$$

$$C_s: A_{95} = 0.05 A_T = 0.05(2.26 \text{ in}^2) = 0.113 \text{ in}^2 = 0.0766 D_e^2$$

$$S_m = 20 \text{ ksi (FIG. 5-8)} \quad D_e = \sqrt{0.113 / 0.0766} = 1.215 \Rightarrow C_s = 0.86$$

$$C_m = 1.0; C_{st} = 1.0; LET R = 0.99 - C_R = 0.81.$$

$$S_m' = (0.86)(0.81)(20) = 13.9 \text{ ksi}; \sigma_m = 5757 \text{ psi}; \sigma_a = 4112 \text{ psi} - \text{PROB. 7.}$$

$$\frac{1}{N} = \frac{5757 \text{ psi}}{58000 \text{ psi}} + \frac{1.0(4112 \text{ psi})}{13900 \text{ psi}} = 0.395; \quad N = 2.53$$

CHECK YIELD:

$$N = \frac{S_y}{K_t(\sigma_a + \sigma_m)} = \frac{36000}{1(4112 + 5757)} = 3.65$$

 $N = 2.53$ IS SATISFACTORY IF NO UNUSUAL CONDITIONS OR UNCERTAINTY OF DATA EXIST.

22.

DATA SAME AS PROBLEM 9, FLUCTUATING NORMAL STRESS - CASE 5.

 $\sigma_m = 386 \text{ MPa}$; $\sigma_a = 89 \text{ MPa}$. FROM APP. 3 AISI 4140 OQT 400 HAS THE HIGHEST S_m WITH $>10\%$ ELONGATION FOR GOOD DUCTILITY.

$$S_u = 2000 \text{ MPa}; S_y = 1730 \text{ MPa}; S_m = 450 \text{ MPa (FIG. 5-8)}; LET R = 0.99 - C_R = 0.81$$

$$S_m' = (0.81)(450 \text{ MPa}) = 364 \text{ MPa} \quad \text{FOR REF.: } D_e = 0.808 \sqrt{12(60)} = 0.808 \sqrt{0.6(5)} = 1.40 \text{ mm}$$

$$\frac{1}{N} = \frac{\sigma_m}{S_m} + \frac{K_t \sigma_a}{S_m'} = \frac{386}{2000} + \frac{(1.0)(89)}{364} = 0.438; \quad N = 2.29$$

SUGGEST TRYING TO FIND AN EVEN STRONGER MATERIAL FROM OTHER REFERENCES. MAY ADJUST WIDTH OR THICKNESS OF SPRING STOCK. USE ANALYSIS FROM PROB. 9 TO COMPUTE FORCE VS. DEFLECTION FOR SPRING. CONSIDER MOVING LATCH PIN FARTHER FROM FIXED END OF SPRING. THIS IS A GOOD SPREAD SHEET PROBLEM.

23.

DATA SAME AS PROB. 6. FLUCTUATING NORMAL STRESS, CASE 5.

AT B: $\sigma_m = 8432 \text{ psi}$, $\sigma_a = 2595 \text{ psi}$. FOR ASTM A500 GRADE B: $S_u = 58 \text{ ksi}$, $S_y = 46 \text{ ksi}$
 AT C: $\sigma_m = 6162 \text{ psi}$, $\sigma_a = 3568 \text{ psi}$. $S_u = 20 \text{ ksi}$; $R = 0.99 \rightarrow C_R = 0.81$; C_s FOR $4 \times 4 \times 1/4$ TUBE
 $A_T = 3.59 \text{ in}^2$; $A_{95} = 0.05(3.59) = 0.180 \text{ in}^2 = 0.0766 \text{ in}^2$; $D_o = 1.53 \text{ in} \rightarrow C_s = 0.84$; $S'_u = (0.84)(0.81)(20) = 13.6 \text{ ksi}$

$$\text{AT C: } \frac{1}{N} = \frac{6162}{58000} + \frac{(1.0)(3568)}{13600} = 0.369; \quad \boxed{N = 2.71} \quad \text{CHECK YIELD: } N = \frac{46000}{1(3429 + 5972)} = 4.92 \text{ OK}$$

AT B: $N = 2.97 > 2.71 \text{ OK}$. $4 \times 4 \times 1/4$ TUBING WEIGHS 8.78 LB/FT . APP 15-14.
 A LIGHTER BEAM CAN BE DESIGNED BY PLACING 4 in SIDE VERTICAL AND
 USING A THINNER WALL. IN APP 15-15, THE SOURCE - JOHNSON
 OFFERS $4 \times 2 \times 0.134$; 5.223 LB/FT . WITH 4 in SIDE VERTICAL, $S_x = 1.58 \text{ in}^3$.
 ORIGINAL $S_x = 1.48$ SO BEAM IS SAFE. WT IS REDUCED BY $\approx 40\%$.

24.

PISTON ROD. FIG. P5-24. DIA = 0.60 in . $A = \pi D^2/4 = 0.283 \text{ in}^2$ FLUCTUATING LOAD. $F_{\max} = 500 \text{ LB TENS.}$; $F_{\min} = -400 \text{ LB COMP.}$ CASE 5.

$$F_m = (500 - 400)/2 = 50 \text{ LB}; \quad F_a = 500 - 50 = 450 \text{ LB}$$

$$\sigma_m = F_m/A = 50 \text{ LB}/0.283 \text{ in}^2 = 174 \text{ psi}; \quad \sigma_a = F_a/A = 450 \text{ LB}/0.283 \text{ in}^2 = 1590 \text{ psi}$$

SAE 4130 WOT 1300; $S_u = 98 \text{ ksi}$; $S_y = 89 \text{ ksi}$; $S_m = 37 \text{ ksi}$ (FIG. 5-8) $C_s = 0.93$; $C_m = 1.0$; $C_{sr} = 0.80$ (AXIAL); $R = 99\%$ - $C_R = 0.81$

$$S'_m = (0.93)(1.0)(0.80)(0.81) 37 \text{ ksi} = 22.3 \text{ ksi}$$

$$\frac{1}{N} = \frac{\sigma_m}{S'_m} + \frac{(K_t)(\sigma_a)}{S'_m} = \frac{174 \text{ psi}}{98000 \text{ psi}} + \frac{(1.0)(1590 \text{ psi})}{22300 \text{ psi}} = 0.170; \quad N = 5.87$$

SAFE BUT HIGH. SHOULD ALSO CHECK FOR K_t IN FINAL DESIGN.IF ROD DIA. IS REDUCED TO 0.50 in . $A = 0.196 \text{ in}^2$

$$\sigma_m = 2801 \text{ psi}; \quad \sigma_a = 4838 \text{ psi}; \quad N = 4.15 \text{ BETTER. USE } D = 0.50 \text{ in}$$

25.

BRITTLE MATERIAL - STATIC LOAD - CASE 1: $N = S_{uc}/\sigma_{\max}$

$$\sigma_{\max} = \frac{K_t F}{A} = \frac{(1.99)(75000 \text{ lb})}{\pi (4.00 \text{ in})^2/4} = 11877 \text{ psi COMPRESSION}$$

$$h/d = 0.25 \text{ in}/4.00 \text{ in} = 0.0625; \quad D/d = 5.00 \text{ in}/4.00 \text{ in} = 1.25; \quad \text{THEN } K_t = 1.99$$

$$N = \frac{S_{uc}}{\sigma_{\max}} = \frac{140000 \text{ psi}}{11877 \text{ psi}} = 11.8$$

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26.

BRITTLE MATERIAL - STATIC LOAD - CASE 1: $N = S_{uc}/\sigma_{\max}$

$$\sigma_{\max} = \frac{K_t F}{A} = \frac{(1.99)(12000 \text{ lb})}{\pi (4.00 \text{ in})^2/4} = 1900 \text{ psi}; \quad [K_t \text{ SAME AS PROB. 25}]$$

$$N = \frac{S_{uc}}{\sigma_{\max}} = \frac{40000 \text{ psi}}{1900 \text{ psi}} = 21.0$$

27.

BRITTLE MATERIAL - BIAxIAL STRESS - SECTION 5-11.1

USE MODIFIED MOHR
METHOD σ_1, σ_2 FOUND FROM MOHR CIRCLE
STRESS ELEMENT IN FILLET AREAAXIAL COMPRESSIVE STRESS FROM PROB.: $\sigma_{\max} = -11877 \text{ psi} = \sigma_y$

CONTINUED - NEXT PAGE.

27.

CONTINUED

TORSION: $T = \frac{K_t T}{Z_P}$; FOR $r/d = 0.0625$; $D/d = 1.25$, $K_t = 1.48$
SEE PROB. 25.

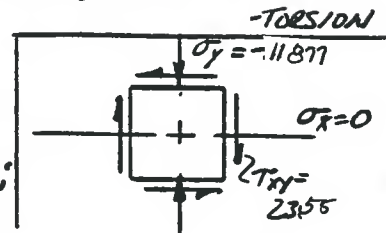
$$Z_P = \frac{\pi D^3}{16} = \frac{\pi (4.00 \text{ IN})^3}{16} = 12.57 \text{ IN}^3$$

$$T = (1.48 \times 20,000 \text{ LB} \cdot \text{IN}) / 12.57 \text{ IN}^3 = 2355 \text{ PSI}$$

FROM MOHR CIRCLE: $R = \sqrt{2355^2 + 5939^2} = 6389 \text{ PSI}$

$$\sigma_1 = \sigma_{AV} + R = -5939 + 6389 = 450 \text{ PSI TENSION}$$

$$\sigma_2 = \sigma_{AV} - R = -5939 - 6389 = -12328 \text{ PSI COMP.$$



GRAPHICAL SOLUTION
4TH QUADRANT

PT. A AT $\sigma_1 = 450 \text{ PSI}$, $\sigma_2 = -12328 \text{ PSI}$

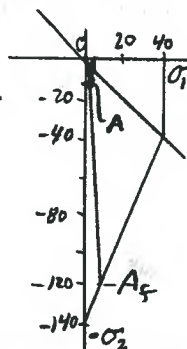
LINE $OA = 12336 \text{ PSI}$

PT A_F IS FAILURE POINT

LINE $OA_F = 120300 \text{ PSI (SCALED)}$

$$N = \frac{OA_F}{OA} = 9.75$$

VERY SAFE



28.

DUCTILE MATERIAL - STATIC LOAD - CASE 2:

SAE 1137 CD ; $S_y = 565 \text{ MPa}$

$$\sigma_B = S_y / N = 565 \text{ MPa} / 3 = 188 \text{ MPa}$$

IN MIDDLE OF SHAFT, $M = 337.5 \text{ kN} \cdot \text{mm}$

$$R_{EQ'D} S = \frac{M}{\sigma_B} = \frac{337.5 \times 10^3 \text{ N} \cdot \text{mm}}{188 \text{ N/mm}^2} = 1795 \text{ mm}^3$$

$$S = \pi D^3 / 32 ; D = \sqrt[3]{32 S / \pi} = 26.3 \text{ mm}$$

USE PREFERRED VALUE $D = 28 \text{ mm}$
(TABLE A2-1)

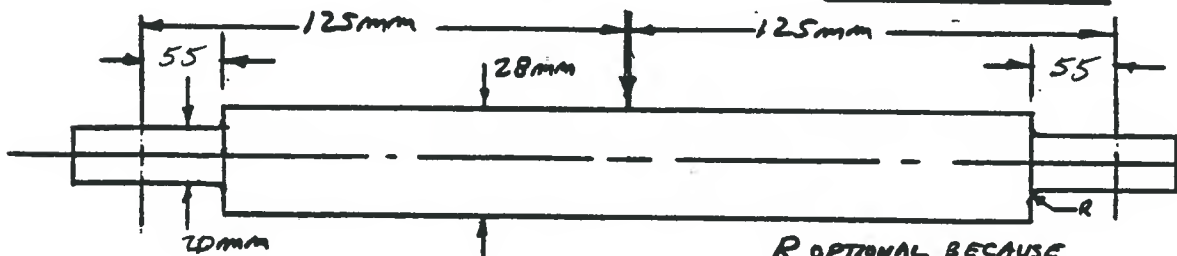
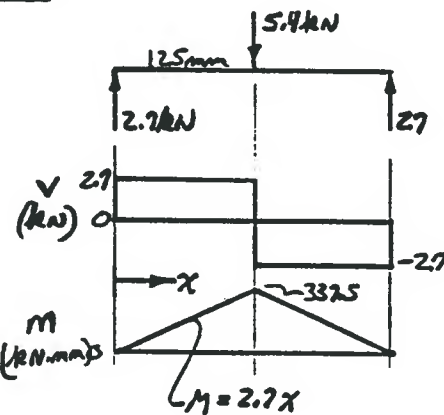
FIND X WHICH WOULD BE SAFE FOR $d = 20 \text{ mm}$ (kN.m)

$$S = \pi d^3 / 32 = \pi (20)^3 / 32 = 785.4 \text{ mm}^3$$

$$M = \sigma_B S = (188 \text{ N/mm}^2) (785.4 \text{ mm}^3) = 147,650 \text{ N} \cdot \text{mm} = 147.7 \text{ kN} \cdot \text{mm}$$

$$\text{BUT } M = 2.7 X ; X = \frac{M}{2.7} = \frac{147.7 \text{ kN} \cdot \text{mm}}{2.7} = 54.7 \text{ mm (MAX)}$$

USE $a = 55 \text{ mm}$



R OPTIONAL BECAUSE
OF STATIC STRESS

29.

FIGURE 5-28. SEE ALSO PROB. 28.

CASE 4; REPEATED REVERSED NORMAL STRESS; $\sigma_d = S_m'/N$ SAE 1137 CD $S_u = 676 \text{ MPa}$; $S_m = 250 \text{ MPa}$ (FIG 5-8); FOR $d = 20 \text{ mm}$,
 $C_s = 0.90$, $C_m = 1.0$, $C_{tr} = 1.0$, $R = 99\%$ - $C_R = 0.81$; $S_m' = (0.90)(0.81)(250) = 182 \text{ MPa}$ $\sigma_d = S_m'/N = 182/3 = 60.7 \text{ MPa}$; $M_{\max} = 337.5 \times 10^3 \text{ N}\cdot\text{mm}$ AT LOAD (PROB. 28)

$$\text{REQ'D } S = \frac{M}{\sigma_d} = \frac{337.5 \times 10^3 \text{ N}\cdot\text{mm}}{60.7 \text{ N/mm}^2} = 5563 \text{ mm}^3; \text{ BUT } S = \pi D^3/32$$

$$\text{REQ'D } D = \sqrt[3]{32 S / \pi} = 38.4 \text{ mm}; \text{ USE } D = 40.0 \text{ mm} \quad \left\{ \begin{array}{l} \text{NEW } C_s = 0.83 \\ \text{RECOMPUTE } D_{\min} = 38.4 \text{ mm} \\ D = 40.0 \text{ mm OK} \end{array} \right.$$

ALLOWABLE DISTANCE "a". LET $R = 2.0 \text{ mm}$, $R/d = 2.0/20 = 0.100$ } $K_t = 1.80$

$$\sigma = \frac{K_t M}{S}; M_{\max} = \frac{\sigma_d S}{K_t}; S = 785.4 \text{ mm}^3 \text{ (PROB. 28)}; D/d = 40/20 = 2.00$$

$$M_{\max} = \frac{60.7 \text{ N/mm}^2 \cdot 785.4 \text{ mm}^3}{1.80} = 26485 \text{ N}\cdot\text{mm} = 26.5 \text{ kN}\cdot\text{mm} = 2.7 \text{ X (PROB. 28)}$$

$$X_{\max} = M_{\max} / 2.7 = 26.5 \text{ kN}\cdot\text{mm} / 2.7 \text{ kN} = 9.81 \text{ mm}; \text{ USE } a = 9.0 \text{ mm (SMALL)}$$

30.

SEE FIG. P5-28 AND PROB. 28 AND 29. $K_t = 2.0$ FOR KEYSEAT MONOCIRCLE FOR ALTERNATING STRESS

$$\text{EQN. 5-22: } \frac{1}{N} = \frac{K_t (\sigma_a)_{\max}}{S_{sm}} + \frac{(\tau_m)_{\max}}{S_{sm}} \quad \left\{ \begin{array}{l} (\tau_m)_{\max} = T/2Z_p \\ (\sigma_a)_{\max} = \sigma_d/2 \end{array} \right.$$

$$\sigma_a = \frac{M_a}{S} = \frac{M_a}{Z_p/2} = \frac{2M_a}{Z_p}; \text{ THEN } (\tau_a)_{\max} = \frac{M_a}{Z_p}$$

$$\frac{1}{N} = \frac{K_t M_a}{Z_p S_{sm}} + \frac{T}{Z_p S_{sm}}; \text{ REQ'D } Z_p = N \left[\frac{K_t M_a}{S_{sm}} + \frac{T}{S_{sm}} \right] \quad \textcircled{I}$$

SAE 1137 CD: $S_u = 676 \text{ MPa}$, $S_m = 250 \text{ MPa}$ ASSUME $D = 50 \text{ mm}$, $C_s = 0.81$, $C_R = 0.81$

$$S_{sm}' = 0.577 S_m' = (0.50)(0.81)(0.81)(250) = 820 \text{ MPa}$$

$$S_{sm} = 0.75 S_u = 0.75(676) = 507 \text{ MPa}$$

$$\text{REQ'D } Z_p = 3 \left[\frac{(2.0)(337.5 \times 10^3)}{820} + \frac{150 \times 10^3}{507} \right] = 25579 \text{ mm}^3 = \pi D^3/16$$

$$D_{\min} = \sqrt[3]{16 Z_p / \pi} = \sqrt[3]{16(25579) / \pi} = 50.69 \text{ mm}; \text{ SPECIFY } D = 52.0 \text{ mm}$$

LOCATION OF STEP FROM 20 mm TO 50 mm:

$$Z_p = \pi D^3/16 = \pi (20)^3/16 = 1571 \text{ mm}^3; \text{ SOLVE EQ } \textcircled{I} \text{ FOR } M_a$$

$$M_a = \frac{S_{sm}}{K_t} \left[\frac{Z_p}{N} - \frac{T}{S_{sm}} \right]$$

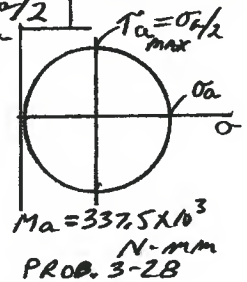
FOR $d = 20 \text{ mm}$; $C_s = 0.90$; $S_{sm}' = (0.90)(0.81)(250)(0.577) = 105 \text{ MPa}$

$$M_a = \frac{105 \text{ N/mm}^2}{1.82} \left[\frac{1571 \text{ mm}^3}{3} - \frac{150 \times 10^3 \text{ N}\cdot\text{mm}}{507 \text{ N/mm}^2} \right] = 13143 \text{ N}\cdot\text{mm}$$

FROM PROB. 3-28, $M = (2700 \text{ N}) X$

$$X_{\max} = M_a / 2700 = \frac{13143 \text{ N}\cdot\text{mm}}{2700 \text{ N}} = 4.86 \text{ mm} \quad \text{VERY SMALL}$$

X IS FROM MIDDLE OF BEARING TO STEP.

REDESIGN IS REQUIRED, CONSIDER LARGER d OR MATERIAL WITH HIGHER STRENGTH.

FOR K_t : BENDING
 $D = 52 \text{ mm}$
 $d = 20 \text{ mm}$
 $r = 2.0 \text{ mm}$
 $K_t = 1.82$

31.

$$S_y = 30 \text{ ksi} ; T_d = 0.5 S_y / 4 = 0.5(30) / 3 = 5 \text{ ksi} = 5000 \text{ psi}$$

CASE 3 : COMBINED STRESS-STATIC LOAD, MAX SHEAR STRESS METHOD.

ASSUME AXIAL COMPRESSION IS SMALL. THEN MAX T OCCURS AT FRONT AND REAR - COMBINED BENDING AND TORSION. USE

EQUIVALENT TORQUE METHOD - CH. 4, EQ. 4-16 AND 4-17

$$T = (200 \text{ LB})(18 \text{ IN}) = 3600 \text{ LB-IN} \quad \text{NOTE: BENDING DUE TO 200 LB LOAD IS ONLY 4000 LB-IN AND ACTS AT A DIFFERENT POINT.}$$

$$M = (400 \text{ LB})(18 \text{ IN}) = 7200 \text{ LB-IN}$$

$$T_e = \sqrt{M^2 + T^2} = \sqrt{7200^2 + 3600^2} = 8050 \text{ LB-IN}$$

$$T_{MAX} = T_e / Z_p = T_d ; \text{ THEN } Z_p = \frac{T_e}{T_d} = \frac{8050 \text{ LB-IN}}{5000 \text{ LB-IN}} = 1.61 \text{ IN}^3$$

$$\text{BUT } S = Z_p / 2 = 0.805 \text{ IN}^3 \rightarrow \text{FROM APP. A-15 USE } 2\frac{1}{2} \text{ IN SCH. 40 PIPE}$$

$$S = 1.06 \text{ IN}^3 ; Z_p = 2.128 \text{ IN}^3 ; A = 1.704 \text{ IN}^2$$

$$\text{CHECK } T_{MAX} = \sqrt{\left(\frac{T}{Z_p}\right)^2 + T^2} = \sqrt{\left(\frac{3600}{2.128}\right)^2 + (1699)^2} = 3903 \text{ psi} < T_d \text{ OK}$$

$$\sigma = \sigma_b - \sigma_c = \frac{-M}{S} - \frac{400 \text{ LB}}{A} = \frac{-7200}{1.064} - \frac{400}{1.704} = -7002 \text{ psi}$$

$$T = \frac{T}{Z_p} = \frac{3600}{2.128} = 1692 \text{ psi}$$

32.

FLUCTUATING SHEAR STRESS:

$$\text{EQN 5-22: } \frac{1}{N} = \frac{T_m}{S_{SM}} + \frac{K_t T_a}{S_{SM}}$$

$$T_m = \frac{T_m}{Z_p} ; T_a = \frac{T_a}{Z_p}$$

$$\frac{1}{N} = \frac{T_m}{Z_p S_{SM}} + \frac{K_t T_a}{Z_p S_{SM}} = \frac{1}{Z_p} \left[\frac{T_m}{S_{SM}} + \frac{K_t T_a}{S_{SM}} \right]$$

$$Z_p = \frac{\pi D^3}{16} = N \left[\frac{T_m}{S_{SM}} + \frac{K_t T_a}{S_{SM}} \right] = 2 \left[\frac{47.5 \times 10^3 \text{ N-mm}}{870 \text{ N/mm}^2} + \frac{2.5(17.5 \times 10^3) \text{ N-mm}}{146 \text{ MPa}} \right]$$

$$S_{SM} = 0.75 S_u = 0.75(1160 \text{ MPa}) = 870 \text{ MPa}$$

USE $S_{SM} = 0.577 S_u$ SAE 4140 OQT 1000; $S_u = 1160 \text{ MPa}$, $S_y = 1056 \text{ MPa}$
 FOR $S_u = 1160 \text{ MPa}$; $S_m = 400 \text{ MPa}$ (FIG 5-8); LET $C_s = 0.90$, $C_R = 0.81$
 $S_{SM} = 0.577(0.9)(0.81)(1160 \text{ MPa}) = 146 \text{ MPa}$

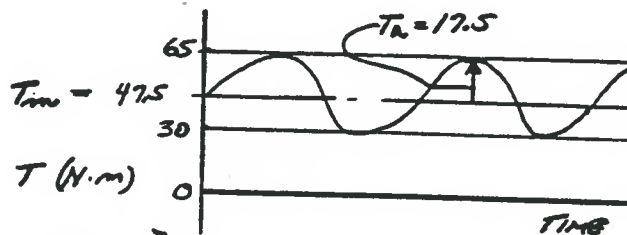
$$\text{REQD } Z_p = 708 \text{ mm}^3 = \pi D^3 / 16$$

$$D_{MIN} = \sqrt[3]{16 Z_p / \pi} = \sqrt[3]{16(708) \text{ mm}^3 / \pi} = 15.31 \text{ mm}$$

CHECK YIELDING: EQN 5-23

$$T_a = \frac{T_a}{Z_p} = \frac{17500 \text{ N-mm}}{804 \text{ mm}^3} = 21.77 \text{ MPa} ; T_m = \frac{T_m}{Z_p} = \frac{47500}{804} = 59.06 \text{ MPa}$$

$$N_f = \frac{S_y / 2}{K_t(T_a + T_m)} = \frac{1056 / 2}{2.5(21.77 + 59.06)} = 2.60 > 2.0 \text{ OK}$$



SPECIFY $D = 16.0 \text{ mm}$
 ACTUAL $C_s = 0.92 \text{ OK}$
 $Z_p = \pi(16)^3 / 16 = 804 \text{ mm}^3$

33.

FLUCTUATING NORMAL STRESS:

$$\text{CASE 5: } \frac{1}{N} = \frac{\sigma_m}{S_m} + \frac{K_t \sigma_a}{S_m'} \quad \text{FIG. 5-8}$$

$$S_y = 58 \text{ KSI}; S_m = 75 \text{ KSI}; N = 3; S_m = 28 \text{ KSI}$$

ASSUME MACHINED SURFACE AND $C_s = 0.9, C_a = 0.81$

$$S_m' = (0.9)(0.81)(28 \text{ KSI}) = 20.4 \text{ KSI} = 20,400 \text{ PSI}$$

$$M_{\max} = \frac{FL}{4} = \frac{800(48)}{4} = 9600 \text{ LB-IN.}$$

$$M_m = M_a = 4800 \text{ LB-IN}$$

$$S = b^3/6; A = b^2$$

$$\sigma_m = \frac{F_x}{A} + \frac{M_m}{S}$$

$$\sigma_m = \frac{1500}{b^2} + \frac{4800}{b^3/6} = \frac{1500}{b^2} + \frac{28800}{b^3}$$

$$\sigma_a = \frac{M_a}{S} = \frac{4800}{b^3/6} = \frac{28800}{b^3}$$

$$\textcircled{1} \quad \frac{1}{N} = \frac{\frac{1500}{b^2} + \frac{28800}{b^3}}{75000} + \frac{\frac{28800}{b^3}}{20,400} = \frac{0.020}{b^2} + \frac{0.384}{b^3} + \frac{1.412}{b^3} = \frac{0.020}{b^2} + \frac{1.796}{b^3}$$

$$\text{TERM INVOLVING } b^2 \text{ IS SMALL: } b \approx \sqrt[3]{N(1.796)} = 1.75 \text{ IN} \rightarrow \text{USE } b = 1.80 \text{ IN.}$$

RECHECK: C_s FOR $b = 1.80 \text{ IN}$ SQUARE. $P_c = 0.808 \sqrt{bh} = 0.808 \sqrt{b^2} = 0.808 b$

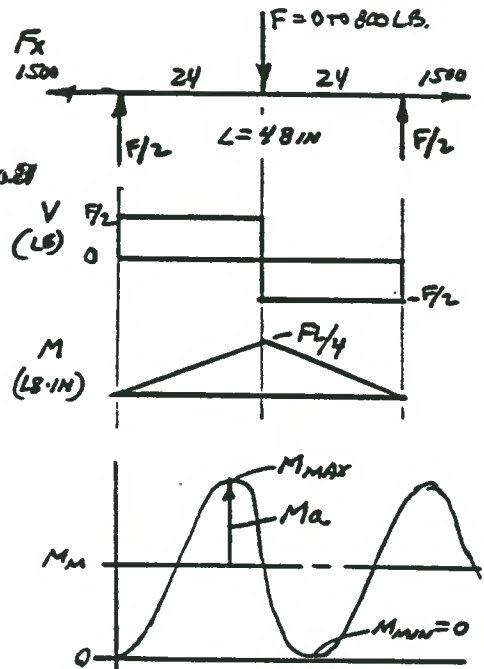
$$P_c = 0.808(1.80) = 1.454 \text{ IN. THEN } C_s = \left(\frac{1.454}{0.3} \right)^{-0.11} = 0.84$$

$$S_m' = (0.84)(0.81)(28 \text{ KSI}) = 19.05 \text{ KSI} = 19,050 \text{ PSI}$$

EQUATION $\textcircled{1}$ BECOMES:

$$\frac{1}{N} = \frac{\frac{1500}{b^2} + \frac{28800}{b^3}}{19,050} + \frac{\frac{28800}{b^3}}{20,400} = \frac{0.020}{b^2} + \frac{0.384}{b^3} + \frac{1.512}{b^3} = \frac{0.020}{b^2} + \frac{1.896}{b^3}$$

$$\frac{1}{N} = \frac{0.020}{(1.80)^2} + \frac{1.896}{(1.80)^3} = 0.331; \quad N = 3.02 \text{ OK} \quad \text{SPECIFY } b = 1.80 \text{ IN}$$



34

FLUCTUATING COMBINED STRESS:

$$\text{CASE 5: } \frac{1}{N} = \frac{(\tau_m)_{\max}}{S_{SM}} + K_t \frac{(\tau_a)_{\max}}{S_{SM}'}$$

T DUE TO TORQUE: (FIG. 3-10)

$$\tau = \frac{T}{\phi} = \frac{T}{0.20865} = \frac{1200}{0.20865} = \frac{5769}{b^3}$$

FROM PROB. 33, MEAN STRESS

$$\sigma_a = \frac{1500}{b^2} + \frac{28800}{b^3} \approx \frac{28800}{b^3}$$

(SMALL)

$$R(\tau_m)_{\max} = \sqrt{\left(\frac{5769}{b^3}\right)^2 + \left(\frac{14400}{b^3}\right)^2} = \frac{15573}{b^3}$$

ALT. STRESS: $\tau_a = \sigma_a/2$ FOR BENDING ONLY

$$(\tau_a)_{\max} = \frac{1}{2} \frac{28800}{b^3} = \frac{14400}{b^3}$$

$$\text{THEN } \frac{1}{N} = \frac{\frac{15573}{b^3}}{56250} + \frac{\frac{14400}{b^3}}{10860} = \frac{1.602}{b^3}$$

FOR $N = 3$, $b = 1.69$ IN \rightarrow USE $b = 1.80$

CHECK:

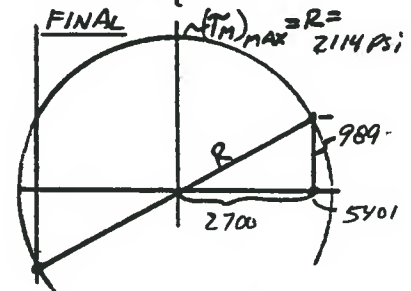
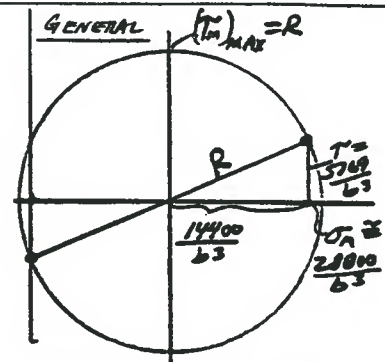
$$\sigma_m = \frac{1500}{(1.80)^2} + \frac{28800}{(1.80)^3} = 5401 \text{ psi}$$

$$\tau_m = \frac{5769}{(1.80)^3} = 989 \text{ psi}$$

$$\tau_a = \frac{14400}{(1.80)^3} = 2469 \text{ psi}$$

$$\frac{1}{N} = \frac{2876}{56250} + \frac{2469}{10860} = 0.279 \rightarrow N = 3.59 \text{ OK}$$

MOHR'S CIRCLE-MEAN STRESS



$$S_{SM} = 0.75 S_u = 0.75(75 \text{ ksi}) = 56.25 \text{ ksi}$$

$$S_{SM}' = 0.577 S_u' = (0.577)(0.83)(28 \text{ ksi}) = 10860 \text{ psi}$$

PROBLEMS 35, 36 AND 37 ALL DUCTILE MATERIALS - STEADY LOAD: CASE 2.

35.

$$\sigma = \frac{F}{A} = \frac{4500 \text{ N}}{\pi(18^2 - 12^2)/4 \text{ mm}^2} = 31.8 \text{ MPa}; N = \frac{S_y}{\sigma} = \frac{290}{31.8} = 9.11$$

SAE 1040 HR; $S_y = 290 \text{ MPa}$

36.

$$\sigma = F/A = \frac{5000 \text{ N}}{(2)^2 \text{ mm}^2} = 34.7 \text{ MPa}; N = \frac{S_y}{\sigma} \text{ CASE 2.}$$

$$a) 1020 \text{ HR}; N = 207/34.7 = 5.96$$

$$b) 8650 \text{ OQT } 1000; N = 1070/34.7 = 30.8$$

$$c) \text{DUCTILE IRON, 60-40-18}; N = 276/34.7 = 7.95$$

$$d) \text{ALUM. 6061-T6}; N = 276/34.7 = 7.95$$

$$e) \text{TI-6AL-4V}; N = 827/34.7 = 23.8$$

$$f) \text{PVC}; N \text{ BASED ON TENSILE STRENGTH}; N = 41/34.7 = 1.18$$

$$g) \text{PHENOLIC}; N = 45/34.7 = 1.30$$

ALL HIGH-REDUCE

LOW-INCREASE

37.

$$\sigma = \frac{F}{A} = \frac{12600 \text{ LB}}{[(2.29)^2 - (2.00)^2] \text{ in}^2} = 11859 \text{ psi}; N = \frac{S_y}{\sigma} = \frac{40000}{11859} = 3.37$$

38 DUCTILE MATERIAL - STEADY LOAD - CASE 2 : PROBS. 38, 39, 40.

SAE 1144 CD $S_y = 90 \text{ KSI} = 90000 \text{ PSI}$

$$A = 15 \cdot 3.5 = 52.5 \text{ IN}^2$$

$$\sum M_B = 0 = 75F - 60C : C = 75(2500)/60 = 3125 \text{ LB} = F_{AC}$$

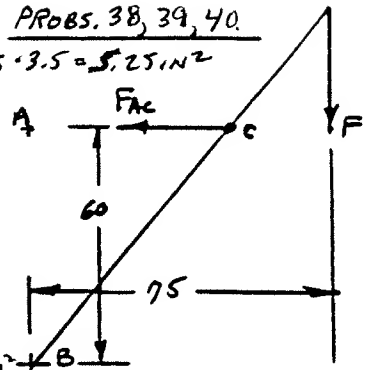
$$\sigma = \frac{F_{AC}}{A} = \frac{3125 \text{ LB}}{(15)(3.5)} = 595 \text{ PSI}$$

$$N = S_y / \sigma = 90000 / 595 = 151 \text{ VERY HIGH}$$

CONSIDER A SMALL ROD OF SAE 1020 HR STEEL

$$\sigma_s = S_y / N = 30000 / 3 = 10000 \text{ PSI} ; A = \frac{F_{AC}}{\sigma_s} = 0.3125 \text{ IN}^2$$

A SQUARE BAR $3/4$ IN ON A SIDE WOULD DO. $A = 0.44 \text{ IN}^2$
OR FROM APP. 15-15: RECT. TUBE - $1.00 \times 2.00 \times 0.065$; $A = 0.373 \text{ IN}^2$



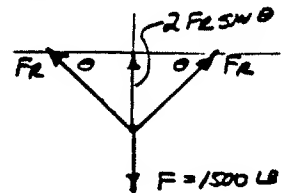
39 $\sum F_v = 0 = 1500 \text{ LB} - 2 F_R \sin 45^\circ$

$$F_R = 1500 / 2 (\sin 45^\circ) = 1061 \text{ LB}$$

$$\sigma_s = S_y / N = 42000 / 3 = 14000 \text{ PSI} = F_R / A$$

$$A = \frac{F_R}{\sigma_s} = \frac{1061 \text{ LB}}{14000 \text{ PSI}} = 0.0758 \text{ IN}^2 = \pi D^2 / 4 : D = 0.31 \text{ IN}$$

USE $9/16$ IN.



40. $F_R = 1500 / 2 (\sin 15^\circ) = 2898 \text{ LB} : A = \frac{F_R}{\sigma_s} = \frac{2898}{14000} = 0.207 \text{ IN}^2 : D = 0.513 \text{ IN}$
USE $9/16$ IN.

41 REPEATED REVERSED AXIAL LOAD: $C_s = 1.00$; $C_{ST} = 0.8$ AXIAL LOAD
 $C_R = 0.81$

CASE 4: $N = S_m' / \sigma_{\max} : \sigma_{\max} = K_t F / A = \frac{1.83(7500 \text{ N})}{(6)(9) \text{ mm}^2} = 254 \text{ MPa}$

$$r/t = 1.5/9 = 0.167 \quad K_t = 1.83$$

$$r_w = 9/12 = 0.75 \quad \text{USING FIG. 3-26(a)}$$

SIZE SAE 4140 DQT 1000

$$S_m = 1160 \text{ MPa} \rightarrow S_m = 400 \text{ MPa (FIG. 5-9)} ; S_m' = (1.00)(0.8)(400)(0.81) = 259 \text{ MPa}$$

$$N = 259 / 254 = 1.02 \text{ LOW} \rightarrow \text{USE LARGER BAR AND/OR STRONGER MATERIAL.}$$

42 REPEATED REVERSED SHEAR STRESS: CASE 4: $C_s = 0.81$, $C_R = 0.81$

$$\tau_{\max} = \frac{T}{\sum p} = \frac{800 \times 10^3 \text{ N} \cdot \text{mm}}{\pi (50 \text{ mm})^3 / 16} = 32.6 \text{ MPa} : SAE 1040 : S_m = 780 \text{ MPa} ; S_m = 280 \text{ MPa}$$

$$S_m' = (0.5)(0.81)(0.81)(280) = 91.9 \text{ MPa}$$

(FIG. 44-1) (FIG. 5-9)

$$S_{sm} = 0.5 S_m'$$

$$N = S_m' / \tau_{\max} = 91.9 / 32.6 = 2.82 \text{ OK}$$

43 REPEATED - ONE DIRECTION SHEAR STRESS: FLUCTUATING SHEAR STRESS

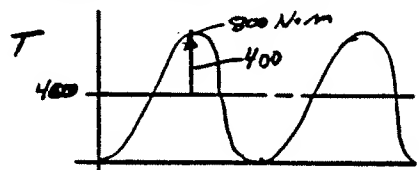
CASE 5: EQ. 5-22: $\frac{1}{N} = \frac{\tau_m}{S_{sm}} + \frac{K_t \tau_a}{S_{sm}'}$

$$\sum p = \pi D^3 / 16 = 24544 \text{ mm}^3$$

$$\tau_m = \tau_a = 400 \times 10^3 \text{ N} \cdot \text{mm} / 24544 \text{ mm}^3 = 16.30 \text{ MPa}$$

$$S_{sm} = 0.75 S_m = 0.75 (780 \text{ MPa}) = 585 \text{ MPa}$$

$$\frac{1}{N} = \frac{16.30}{585} + \frac{(1.0)(16.30)}{91.9} = 0.205 : N = 4.87 \quad (S_{sm}' \text{ FROM PROB. 42})$$



44.

DUCTILE MATERIAL - STATIC LOAD - CASE 2 : $N = 0.5 S_y / T_{MAX}$

$$T_{MAX} = \frac{T}{Z_p} = \frac{88.0 \text{ LB-IN}}{\pi (0.40 \text{ IN})^3 / 6} = 7003 \text{ PSI}$$

LET $N = 3.0$ DESIGN DECISION

$$\text{REQ'D } S_y = N T_{MAX} / 0.5 = 3(7003) / 0.5 = 42018 \text{ PSI}$$

ALUMINUM 2024-T4 HAS $S_y = 47000 \text{ PSI}$

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FLUCTUATING SHEAR STRESS: CASE 5

ASSUME $C_s = 0.82$ $C_R = 0.81$ $N = 3$

$$T_{MAX} = \frac{63000 \text{ (HP)}}{n} = \frac{63000 (110)}{560} = 12375 \text{ LB-IN.}$$

$$T_A = T_{AV} = T_{MAX} / 2 = 6188 \text{ LB-IN.}; T_m = T_{AV} = \frac{T}{2}$$

$$S_{SM} = 0.75 S_u = 0.75 (208) = 156 \text{ KSI}; S_u = 208 \text{ KSI} \rightarrow S_m = 64 \text{ KSI (FIG. 5-8)}$$

$$S_{SM}' = (0.50)(0.82)(0.81)(64) = 21.25 \text{ KSI} = 21250 \text{ PSI}$$

$$\frac{1}{N} = \frac{T_m}{Z_p S_{SM}} + \frac{T_A}{Z_p S_{SM}'} = \frac{1}{Z_p} \left[\frac{T_m}{S_{SM}} + \frac{T_A}{S_{SM}'} \right] : Z_p = N \left[\frac{T_m}{S_{SM}} + \frac{T_A}{S_{SM}'} \right]$$

$$Z_p = 3 \left[\frac{6188}{156000} + \frac{6188}{21250} \right] = 0.993 \text{ IN}^3; D = \sqrt[3]{6 Z_p / \pi} = 1.72 \text{ IN. USC D} = 1.75 \text{ IN.}$$

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STEADY LOAD - CASE 3 : $N = 0.5 S_y / T_{MAX}$

$$T = \frac{P}{n} = \frac{28 \times 10^3 \text{ N-mm/s}}{45 \text{ RAD/s}} = 622 \text{ N-mm} = 622 \times 10^3 \text{ N-mm}$$

$$Z_p = \frac{\pi (D^4 - d^4)}{16 D} = \frac{\pi (40^4 - 30^4)}{16 (40)} \text{ mm}^3 = 8590 \text{ mm}^3$$

$$T_{MAX} = T / Z_p = 622 \times 10^3 \text{ N-mm} / 8590 \text{ mm}^3 = 72.4 \text{ MPa}$$

$$N = 0.5 S_y / T_{MAX} : S_y = N T_{MAX} / 0.5 = 3(72.4) / 0.5 = 434 \text{ MPa}$$

AISI 1040 COLD DRAWN HAS $S_y = 490 \text{ MPa}$.

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FLUCTUATING SHEAR STRESS - CASE 5: EQ. 5-22

(SEE PROBLEM 46.)

$$T_m = P_m / n = 21500 / 45 = 478 \text{ N-mm}$$

$$T_A = P_A / n = 6500 / 45 = 144 \text{ N-mm}$$

$$T_m = T_m / Z_p = 478000 / 8590 = 55.6 \text{ MPa}$$

$$T_A = T_A / Z_p = 144000 / 8590 = 16.8 \text{ MPa}$$

$$\frac{1}{N} = \frac{T_m}{S_{SM}} + \frac{K_t T_A}{S_{SM}'} = \frac{55.6}{S_{SM}} + \frac{(1.0)(16.8)}{S_{SM}'}$$

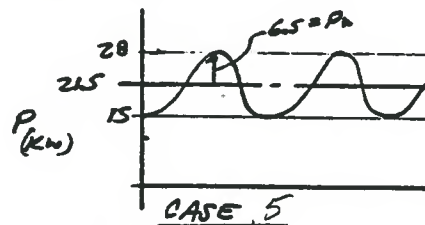
 $C_s = 0.83$ $C_R = 0.81$

AFTER TRIALS:

$$\text{AISI 1144 C.D.}; S_y = 621 \text{ MPa}; S_u = 690 \text{ MPa} \rightarrow S_m = 253 \text{ MPa (FIG. 5-8)}$$

$$S_{SM} = 0.75 S_u = 0.75 (490) = 518 \text{ MPa}; S_{SM}' = 0.50 S_u' = (0.50)(0.83)(0.81)(253) = 85.0 \text{ MPa}$$

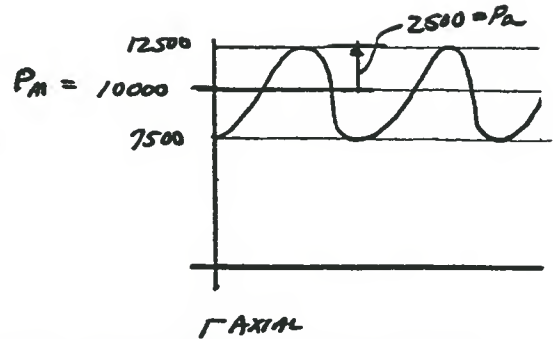
$$\frac{1}{N} = \frac{55.6}{518} + \frac{16.8}{85.0} = 0.305; N = 1 / 0.305 = 3.28 \text{ OK}$$



FLUCTUATING NORMAL STRESS: CASE 5: EQ. 5-20
 PROBLEMS 48, 49, 50, 51.

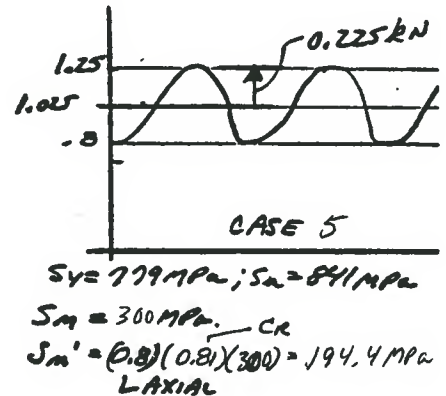
48

CASE 5: $\frac{1}{N} = \frac{\sigma_m}{S_m} + \frac{k_t \sigma_a}{S_m'}$
 $A = (W-d)t = (1.50 - 0.50)(0.50) = 0.50 \text{ in}^2$
 $\sigma_m = \frac{10000}{.5} = 20000 \text{ psi}$
 $\sigma_a = \frac{2500}{0.5} = 5000 \text{ psi}$
 $d/w = .5/1.50 = 0.333 \rightarrow k_t = 2.31$
 $S_y = 107 \text{ ksi}; S_u = 118 \text{ ksi} \rightarrow S_m = 43 \text{ ksi}; S_m' = (0.8)(0.81)(43) = 27.9 \text{ ksi}$
 $\frac{1}{N} = \frac{20000}{118000} + \frac{2.31(5000)}{27900} = 0.583 \rightarrow N = 1.71 \text{ (Low)}$



49

$A = \pi(6)^2/4 = 28.27 \text{ mm}^2$
 $\sigma_m = \frac{F_m}{A} = \frac{1025 \text{ N}}{28.27 \text{ mm}^2} = 36.25 \text{ MPa}$
 $\sigma_a = \frac{F_a}{A} = \frac{225 \text{ N}}{28.27 \text{ mm}^2} = 7.96 \text{ MPa}$
 $r_h = 0.5/6 = 0.083 \}$ $k_t = 2.03$
 $d/d = 9/6 = 1.50$
 $\frac{1}{N} = \frac{36.25}{841} + \frac{(2.03)(7.96)}{194.4} = 0.128; N = 7.92$
 FOR YIELD $N = 8.68 \text{ OK}$
 COULD USE WEAKER MATERIAL

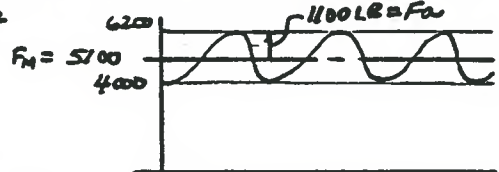


50

FROM PROB. 62, CHAPTER 3, MAX STRESS OCCURS AT BOTTOM
 $k_t = 1.86; S_y = 86000 \text{ psi}; S_u = 121000 \text{ psi}; S_m = 43000 \text{ psi}$
 $S_m' = (0.8)(81)(43000) = 27864 \text{ psi}; \sigma_m = \frac{F_m}{A} = \frac{600 \text{ LB}}{\pi(0.5)^2/4} = 3056 \text{ psi}$
 $\sigma_a = \sigma_m$
 CASE 5:
 $\frac{1}{N} = \frac{\sigma_m}{S_m} + \frac{k_t \sigma_a}{S_m'} = \frac{3056}{121000} + \frac{(1.86)(3056)}{27864} = 0.224; N = 4.56 \text{ OK}$
 FOR YIELD: $N = 7.56 \text{ OK}$

51

FROM PROB. 63, CHAPTER 3, MAX STRESS OCCURS AT LEFT HOLE (0.72 DIA)
 $k_t = 2.15; A = (1.40 - 0.72)(0.50) = 0.34 \text{ in}^2$
 $\sigma_m = \frac{F_m}{A} = \frac{5100}{.34} = 15000 \text{ psi}$
 $\sigma_a = \frac{F_a}{A} = \frac{1100}{.34} = 3235 \text{ psi}$
 CASE 5:
 $\frac{1}{N} = \frac{\sigma_m}{S_m} + \frac{k_t \sigma_a}{S_m'} = \frac{15000}{145000} + \frac{2.15(3235)}{33050} = 0.314$
 $N = 3.19$
 FOR YIELD: $N = \frac{S_y}{k_t(\sigma_a + \sigma_m)} = \frac{125000}{2.15(3235 + 15000)} = 3.19 \text{ O.K.}$



$S_y = 125 \text{ ksi}$
 $S_u = 145 \text{ ksi}$
 $S_m = 51 \text{ ksi}$
 $S_m' = (0.8)(0.81)(51) = 33.05 \text{ ksi}$
 CSR CR

52

FROM PROB 3-64, $\sigma_{max} = 16650 \text{ PSI}$ INCLUDING K_t : CASE 1: $N = S_{ut}/\sigma$
REQ'D $S_{ut} = N\sigma = 3(18281) = 54843 \text{ PSI} \rightarrow \underline{\text{USE GRADE 60 CAST IRON}}$

53

CASE 5: $\frac{1}{N} = \frac{\sigma_m}{S_u} + \frac{K_t \sigma_a}{S_m'}$: NOTE THAT A DIRECT SOLUTION IS NOT POSSIBLE BECAUSE BOTH S_u AND S_m' ARE UNKNOWN. ALSO DATA FOR ENDURANCE FOR TITANIUM ARE NOT DIRECTLY AVAILABLE HERE. AS AN ESTIMATE WE WILL USE FIG 5-8 AND THE DISCUSSION FOR STEEL TO OBTAIN S_m' . ALSO NOTE FROM PREVIOUS PROBLEMS, $S_m' \approx S_u/4$. THIS PERMITS SOLUTION FOR S_u . AFTER MATERIAL SELECTION, FINAL "N" CAN BE COMPUTED.

FROM PROBLEM 3-65, $K_t = 2.30$

$$A = \pi(30)^2/4 = 707 \text{ mm}^2$$

$$\sigma_m = \frac{F_m}{A} = \frac{25.15 \times 10^3 \text{ N}}{707 \text{ mm}^2} = 35.57 \text{ MPa}$$

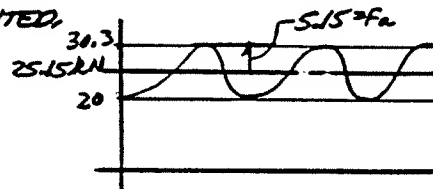
$$\sigma_a = \frac{F_a}{A} = \frac{5.15 \times 10^3 \text{ N}}{707 \text{ mm}^2} = 7.28 \text{ MPa}$$

$$\frac{1}{N} = \frac{\sigma_m}{S_u} + \frac{K_t \sigma_a}{S_m'} = \frac{35.57}{S_u} + \frac{(2.30)(7.28)}{S_u/4} = \frac{102.5}{S_u}$$

THEN $S_u = N(102.5) \approx 3(102.5) = 308 \text{ MPa}$: TRY Ti 50A; $S_y = 276 \text{ MPa}$; $S_m = 345 \text{ MPa}$
FROM FIG. 5-8: $S_m \approx 120 \text{ MPa}$ (ESTIMATE)

$$S_m' = (0.8)(0.81)(120) = 84.2 \text{ MPa}$$

$$\frac{1}{N} = \frac{35.57}{345} + \frac{(2.30)(7.28)}{84.2} = 0.302 \rightarrow N = 3.31 \quad \underline{\text{OK}} \quad \underline{\text{Ti 50A}}$$



54

FROM PROB. 3-66, $K_t = 1.43$: $T_m = 1100 \text{ LB} \cdot \text{IN} = T_a$

$$Z_p = \pi(1.25)^3/16 = 0.383 \text{ in}^3$$

$$T_m = T_a/Z_p = 1100/0.383 = 2868 \text{ PSI} = T_a \rightarrow \text{CASE 5}$$

$$\frac{1}{N} = \frac{T_m}{S_{su}} + \frac{K_t T_a}{S_{sm}'}$$
 ASSUME $S_{sm}' \approx S_{su}/4$ [SEE PROB 53]

$$\frac{1}{N} = \frac{2868}{S_{su}} + \frac{1.43(2868)}{S_{su}/4} = \frac{19273}{S_{su}}: S_{su} \approx 309273 \approx 57819 \text{ psi}$$

$$\text{BUT } S_{su} = 0.75 S_u; S_u = S_{su}/0.75 = 57819/0.75 = 77092 \text{ PSI}$$

$$\text{TRY AISI 1137 OQT 1300; } S_y = 60 \text{ KSI; } S_u = 87 \text{ KSI; } S_m = 33 \text{ KSI}$$

$$S_{sm}' = 0.50 S_m': S_{sm}' = (0.50)(0.81)(0.85)(33000) = 11360 \text{ psi}$$

$$S_{su} = 0.75 S_u = 0.75(87000 \text{ PSI}) = 65250 \text{ PSI}$$

$$\frac{1}{N} = \frac{2868}{65250} + \frac{1.43(2868)}{11360} = 0.405; N = 2.47 \text{ LOW}$$

$$\text{TRY SAE 1046 WQT 1000; } S_u = 113 \text{ KSI, } S_y = 88 \text{ KSI, } S_m = 42 \text{ KSI}$$

$$S_{su} = 0.75(113) = 84.75 \text{ KSI; } S_{sm}' = (0.50)(0.81)(0.85)(42) = 14.458 \text{ KSI}$$

$$\frac{1}{N} = \frac{2868}{84750} + \frac{1.43(2868)}{14458} = 0.3175 \quad N = 3.15 \quad \underline{\text{OK}}$$

55. USE CASE 1 BECAUSE HIGHER STRENGTH DUCTILE IRONS ARE FAIRLY BRITTLE.
FROM PROB. 3-67, $\sigma = 32564 \text{ psi}$ INCLUDING K_t : $N = S_{ut}/\sigma$
REQ'D. $S_{ut} = N\sigma = 3(32564) = 97692 \text{ psi}$ → USE GRADE 100-70-03.

56. FROM PROB. 3-68, $\sigma = 49323 \text{ psi}$ INCLUDING K_t : CASE 4: $N = S_{ut}'/\sigma$
REQ'D $S_{ut}' = N\sigma = 3(49323) = 147969 \text{ psi}$
REFERRING TO FIG. 5-8, THIS IS VERY HIGH. NO PRACTICAL MATERIAL
REDSIGN THE MEMBER.

57. LOAD IS REPEATED - ONE DIRECTION - TORSIONAL SHEAR STRESS;
FLUCTUATING SHEAR STRESS - CASE 5: $T_m = T_a = 100 \text{ LB} \cdot \text{IN} / 2 = 50 \text{ LB} \cdot \text{IN}$.
AT FILLET: $r/d = 0.50/0.30 = 1.67$; $A/d = 0.025/0.30 = 0.083$; $K_t = 1.43$
 $Z_P = \pi d^3/16 = \pi(0.30)^3/16 = 0.00530 \text{ IN}^3$
 $T_m = T_a = \frac{T}{Z_P} = \frac{50 \text{ LB} \cdot \text{IN}}{0.00530 \text{ IN}^3} = 9431 \text{ psi}$
SAE 8740 OQT 1000: $S_y = 167 \text{ ksi}$; $S_u = 175 \text{ ksi}$; THEN $S_m \approx 60 \text{ ksi}$ (FIG. 5-8)
 $S_{sm}' = (0.50)S_m = (0.50)(0.81)(1.0)60 \text{ ksi} = 24.3 \text{ ksi}$; $S_{su} = 0.75S_u = 0.75(175) = 131.3 \text{ ksi}$
 $\frac{1}{N} = \frac{T_m}{S_{su}} + \frac{K_t T_a}{S_{sm}'} = \frac{9431}{131300} + \frac{1.43(9431)}{24360} = 0.627$; $N = 1.60$ LOW

58. STEADY LOAD - BRITTLE MAT'L.: CASE 1
AT MIDDLE - BETWEEN C AND D:

$$Z = bh^2/6 = (0.75 \times 2.25)^2/6 = 0.633 \text{ IN}^3$$

$$\sigma = M/Z = 2250 \text{ LB} \cdot \text{IN} / 0.633 \text{ IN}^3 = 3556 \text{ psi}$$

AT STEP - POINT B:

$$Z = bh^2/6 = (0.75 \times 1.25)^2/6 = 0.1953 \text{ IN}^3$$

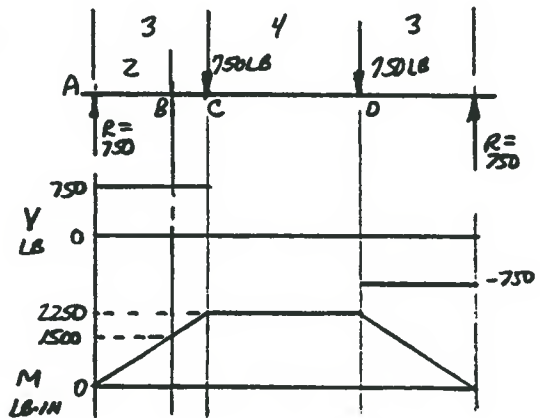
$$r/h = 0.20/1.25 = 0.16$$

$$W/h = 2.25/1.25 = 1.80$$

$$\left. \begin{array}{l} r/h = 0.16 \\ W/h = 1.80 \end{array} \right\} K_t = 1.63$$

$$\sigma_{max} = \frac{K_t M}{Z} = \frac{1.63(1500)}{0.1953} = 12519$$

$$N = S_{ut}/\sigma_{max} = 40000/12519 = 3.19$$



59. REPEATED ONE DIRECTION: CASE 5: EQ. 5-20: $5 \leq N \leq 3.0$

$$D = 0.50 \text{ IN.}; A = \pi D^2/4 = 0.196 \text{ IN}^2$$

$$\text{ASSUME } K_t = 1.0$$

$$\sigma_m = \sigma_a = F/A = 1500 \text{ LB} / 0.196 \text{ IN}^2 = 7639 \text{ psi}$$

1ST TRIAL: SAE 1040 CD: $S_y = 71 \text{ ksi}$; $S_u = 80 \text{ ksi}$

$$S_m \approx 30 \text{ ksi (FIG. 5-8)}; S_{m}' = (C_R)(C_S)S_m = (0.81)(0.8)(30) = 19.4 \text{ ksi}$$

$$\frac{1}{N} = \frac{\sigma_m}{S_u} + \frac{K_t \sigma_a}{S_{m}'} = \frac{7639}{80000} + \frac{1.0(7639)}{19400} = 0.489$$

$$N = 2.04 \text{ LOW}$$

2ND: SAE 1040 WQT 1000: $S_{su} = 112 \text{ ksi}$; $S_y = 87 \text{ ksi}$; 23% ELI; $S_m = 42 \text{ ksi}$; $S_{m}' = 27.2 \text{ ksi}$

$$\frac{1}{N} = \frac{7639}{112000} + \frac{(1)(7639)}{27200} = 0.349$$

$$N = 2.86 \text{ OK}$$

$$\text{CHECK YIELD: EQ. 5-21}$$

$$N = \frac{S_y}{K_t(\sigma_a + \sigma_m)} = \frac{87000}{1(7639 + 7639)} = 5.69 \text{ OK}$$

60

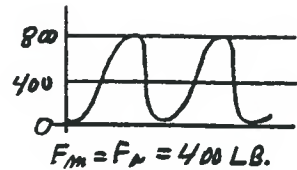
REPEATED - ONE DIRECTION: CASE 5; SPECIFY A STEEL

$$F_{\max} = 800 \text{ LB}; F_{\min} = 0; \delta_{\max} = 0.010 \text{ IN}; L = 25.0 \text{ IN.}$$

CONSIDER DEFLECTION FIRST:

$$\text{REQ'D. } A = PL/ES = \frac{(800)(25.0)}{(30 \times 10^6)(0.010)} = 0.0667 \text{ IN}^2 = S^2$$

$$S = \sqrt{A} = \sqrt{0.0667 \text{ IN}^2} = 0.258 \text{ IN.}; \text{TRY } S = 0.300 \text{ IN.}$$

STRESS ANALYSIS: ASSUME $K_t = 1.0$

$$\sigma_m = \sigma_a = \frac{F}{A} = \frac{400 \text{ LB}}{(0.30 \text{ IN})^2} = 4444 \text{ PSI}$$

TRY SAE 1040 CD: $S_y = 71 \text{ KSI}; S_u = 80 \text{ KSI}; 12\% \text{ ELONGATION}$ FROM FIG 5-8: $S_m = 30 \text{ KSI};$ LET $C_s = 1.0; C_{ST} = 0.80 (\text{AXIAL}); C_R = 0.81$

$$S_m' = (1.0)(0.80)(0.81)(30) = 19,400 \text{ PSI}$$

$$\text{EQ. 5-20: } \frac{1}{N} = \frac{\sigma_m}{S_u} + \frac{K_t \sigma_a}{S_m'} = \frac{4444}{80,000} + \frac{1.0(4444)}{19,400} = 0.285; N = 3.51 \text{ OK.}$$

61

REPEATED - ONE DIRECTION: CASE 5; $F_{\max} = 1200 \text{ LB}; F_{\min} = F_a = 600 \text{ LB.}$

$$a) \frac{1}{N} = \frac{\sigma_m}{S_y} + \frac{K_t \sigma_a}{S_m'}$$

FOR ILLUSTRATION USE SAME MATERIAL

AS IN PROB. 60: SAE 1040 CD

$$S_u = 80 \text{ KSI}; S_m = 30 \text{ KSI}$$

$$S_m' = (0.90)(1.0)(0.81)(30) = 21.9 \text{ KSI}$$

LARGE L BENDING

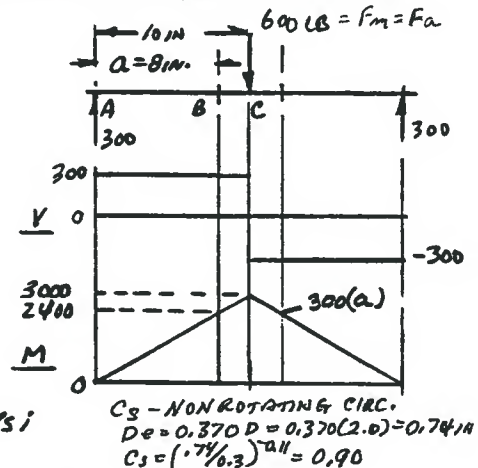
$$\text{AT C: } M = 3000 \text{ LB}\cdot\text{IN}; K_t = 1.0$$

$$S = \pi D^3/32 = \pi (2.0)^3/32 = 0.785 \text{ IN}^3$$

$$\text{EQ. 5-20: } \frac{1}{N} = \frac{\sigma_m}{S_u} + \frac{K_t \sigma_a}{S_m'}$$

$$\sigma_m = \sigma_a = M/S = 3000 \text{ LB}\cdot\text{IN}/0.785 \text{ IN}^3 = 3820 \text{ PSI}$$

$$\frac{1}{N} = \frac{3820}{80,000} + \frac{1.0(3820)}{21,900} = 0.222; N = 4.50 \text{ HIGHER THAN (b).}$$

b) AT SECTION B: $M = 2400 \text{ LB}\cdot\text{IN}; L/d = 0.20/2.0 = 0.10; D/d = 3.0/2.0 = 1.50$

$$K_t = 1.74; \sigma_m = \sigma_a = \frac{M}{S} = \frac{2400}{0.785} = 3056 \text{ PSI}$$

$$\frac{1}{N} = \frac{3056}{80,000} + \frac{(1.74)(3056)}{21,900} = 0.281; N = 3.56 \text{ - LOWER THAN (a).}$$

62

REDESIGN BEAM IN PROB. 61. NOTE THAT N IS INVERSELYPROPORTIONAL TO MOMENT M AND DISTANCE L . THEN L MUST BEREDUCED BY: $L' = L \times 3.51/4.50 = 0.787(L) = 0.787(8.0) = 6.30 \text{ IN. (SAY 6.25 IN)}$ THEN $M = 300(L) = 1875 \text{ LB}\cdot\text{IN.}; \sigma_m = \sigma_a = M/S = 1875/0.785 = 2389 \text{ PSI}$

$$\frac{1}{N} = \frac{2389}{80,000} + \frac{1.74(2389)}{21,900} = 0.220; N = 4.55 \text{ OK HIGHER THAN (a).}$$

SPECIFY $L = 6.25 \text{ IN.}$

63

REFER TO PROBS. 61, 62. : NEW $r = 0.40$; $r/d = 0.40/2.0 = 0.20$

$$D/d = 3.0/2.0 = 1.50 ; K_t = 1.47$$

IF $d = 2.00$ IN AS GIVEN; $M = 2400$ LB-IN AT B; $\sigma_m = \sigma_a = 3056$ PSI

$$\frac{1}{N} = \frac{\sigma_m}{S_u} + \frac{K_t \sigma_a}{S_m'} = \frac{3056}{80000} + \frac{1.47(3056)}{21900} = 0.305 ; N = 3.27$$

DIMENSION d MUST BE REDUCED TO GET $N \geq 4.50$ AS AT C.

$$d' = d \times 4.2/4.50 = 2.0(0.933) = 1.87$$
 IN ; LET $d = 1.75$ IN.

THEN $M = (7.25)(300) = 2175$ LB-IN; $\sigma = M/S = 2771$ PSI

$$\frac{1}{N} = \frac{2771}{80000} + \frac{(1.47)(2771)}{21900} = 0.2206 ; N = 4.53 \text{ OK}$$

64.

REPEATED - ONE DIRECTION; FLUCTUATING STRESS: CASE 5, EQ 5-20

SAE 1040 HR: $S_y = 42$ KSI; $S_u = 72$ KSI; $S_m = 23$ KSI FIG 5-8 HOT ROLLED.

$C_s = 1.0$ DIRECT TENSION; $C_a = 0.8$; $C_{sc} = 0.80$ AXIAL LOAD

$$S_m' = (0.8)(0.8)(23 \text{ KSI}) = 14.9 \text{ KSI}$$

a) AT PIN HOLE: $d = 0.25$ IN. DIA., $w = 1.00$ IN.

$$r/w = 0.25 ; K_t = 4.40$$

$$\sigma_{nom} = \frac{F}{(w-d)t} = \frac{2500 \text{ LB}}{(1.0 - 0.25)(0.25) \text{ IN}^2} = 13333 \text{ PSI}$$

$$\frac{1}{N} = \frac{\sigma_m}{S_u} + \frac{K_t \sigma_a}{S_m'} = \frac{13333}{72000} + \frac{(4.40)(13333)}{19900} = 4.12 ; N = 0.243$$

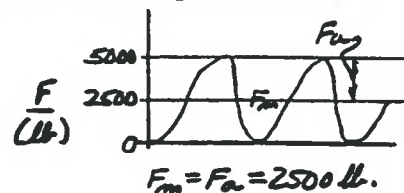
INDICATES FAILURE.

b) AT FILLETS: $r/h = 0.02/1.00 = 0.02$; $H/h = 2.0/1.0 = 2.0$

ON EPATIGUE: MIN. $H/h = 0.04$ THEN $K_t = 3.68$

$$\sigma_{nom} = \frac{F}{A} = \frac{2500 \text{ LB}}{(1.00)(0.25) \text{ IN}^2} = 10000 \text{ PSI}$$

$$\frac{1}{N} = \frac{10000}{72000} + \frac{(3.68)(10000)}{14900} = 2.61 ; N = 0.383 \text{ FAILURE.}$$



65

IMPROVEMENTS: 1.) INCREASE THICKNESS, 2.) INCREASE FILLET RADIUS, 3.) USE STRONGER MATERIAL, 4.) INCREASE PIN HOLE SIZE - OR - CHANGE MANNER OF APPLYING FORCE TO THE PART TO ELIMINATE HOLE - OR - MAKE PART THICKER AT THE HOLES THAN IN MIDDLE OF THE PART. MATERIAL MAY BE REMOVED IN 2.00 IN. SECTION NEAR MIDDLE OF PART TO OFFSET ADDED MATERIAL ELSEWHERE. COULD TRY TITANIUM WITH LOWER DENSITY THAN STEEL. THIS PROBLEM MAY BE TOO RESTRICTIVE TO PERMIT A PRACTICAL SOLUTION WITH DATA IN THIS BOOK. MAY HAVE TO ACCEPT LOWER $N < 3.0$ OR SOME INCREASE IN WEIGHT OF THE COMPONENT.

66

FLUCTUATING NORMAL STRESS - CASE 5 - EQ. 5-20.

SAE 1040 CD: $S_y = 71 \text{ ksi}$; $S_u = 80 \text{ ksi}$; $S_m = 30 \text{ ksi}$ FIG. 5-8: $C_R = 0.81$ $C_s = 1.0$, $C_{st} = 0.80$ AXIAL: $S_m' = 0.8(0.81)(30) = 19.400 \text{ ksi}$ $F_m = (24.8 + 3.0)/2 = 13.9 \text{ kN} (1.0 \text{ lb}/4.448 \text{ N}) = 3125 \text{ LB}$ $F_a = 24.8 - 13.9 = 10.9 \text{ kN} (1.0 \text{ lb}/4.448 \text{ N}) = 2450 \text{ LB}$ $t = 0.375 \text{ in.}$; $w = 1.50 \text{ in.}$; $d = 0.625 \text{ in.}$; $d/w = 0.417$ AT PIN HOLES: $K_t = 2.84$

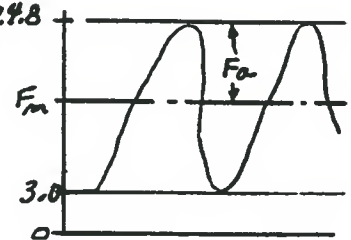
$$\sigma_m = \frac{F_m}{(w-d)t} = \frac{3125}{(1.50-0.625)(0.375)} = 9524 \text{ psi}$$

$$\sigma_a = \frac{F_a}{(w-d)t} = \frac{2450}{(1.50-0.625)(0.375)} = 7467 \text{ psi}$$

$$\frac{1}{N} = \frac{\sigma_m}{S_m} + \frac{K_t \sigma_a}{S_m'} = \frac{9524}{80000} + \frac{(2.84)(7467)}{19400} = 1.212; N = 0.825 \text{ FAILURE.}$$

AT MIDDLE HOLE: $K_t = 2.22$; SAME NOMINAL STRESSES.

$$\frac{1}{N} = \frac{9524}{80000} + \frac{(2.22)(7467)}{19400} = 0.854; N = 1.17 \text{ LOW.}$$

PART MUST BE REDESIGNED.

67

REPEATED REVERSED LOAD - BENDING - CASE 4: $N = S_m' / \sigma_{max}$ SAE 1340 OQT 1300: $S_y = 517 \text{ MPa}$; $S_u = 690 \text{ MPa}$; $S_m = 260 \text{ MPa}$; $C_R = 0.81$ $C_s = 0.98$, $C_{st} = 1.0$, $C_m = 1.0$: $S_m' = (0.98)(0.81)(260) = 206 \text{ MPa}$ $R_F = 400 \text{ N} (250/400) = 250 \text{ N}$ C_s AT B: $D = 0.808(12) = 9.70 \text{ mm}$ $R_A = 400 \text{ N} (150/400) = 150 \text{ N}$ $C_s = 0.98$ AT B: $S = S^3/6 = 12^3/6 = 288 \text{ mm}^3$ $M/h = 2.0/12.0 = 0.167$; $M/h = 2.0/12 = 0.167$ $K_t = 1.60$

$$\sigma_{max} = \frac{K_t M}{S} = \frac{(1.60)(2250)}{288} = 125 \text{ MPa}$$

$$N = S_m' / \sigma_{max} = 206/125 = 1.648$$

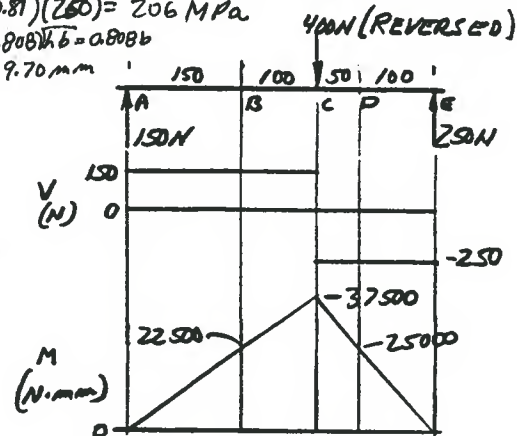
AT C: $S = bh^2/6 = 12(20)^2/6 = 800 \text{ mm}^3$

$$\sigma_{max} = \frac{37500(1.0)}{800} = 46.9 \text{ MPa (AT C.)}$$

AT D: FROM EFANAVE: $d/w = 14/20 = 0.70$ - $K_t = 1.40$

$$\sigma_{min} = \frac{6 M w}{(w^2 - d^2)(t)} = \frac{6(25000)(20)}{(20^2 - 14^2)(12)} = 47.6 \text{ MPa}; \sigma_{max} = K_t \sigma_{min}$$

$$\sigma_{max} = K_t \sigma_{min} = (1.40)(47.6) = 66.6 \text{ MPa (AT D.)}$$

MINIMUM N = 1.648 AT B (LOW)

68

SEE PROB 67: FOR $N = 2.5$; $S_m' = N \sigma_{max} = 2.5(125) = 312.5 \text{ MPa} = (0.98)(0.81) S_m$ THEN S_u REQ'D = 2.1150 MPa

(FIG 5-8)

 $S_m = 394 \text{ MPa}$ FROM APPA 4-3: SAE 1340 OQT 900 HAS $S_u = 1150 \text{ MPa}$

69. FLUCTUATING NORMAL STRESS - CASE 5: $F_{min} = 300 \text{ LB}$; $F_{max} = 700 \text{ LB}$

$$F_m = (300 + 700)/2 = 500 \text{ LB}; F_a = 700 - 500 = 200 \text{ LB}$$

$$\text{AT FILLET: } r/d = 0.06/1.25 = 0.048; D/d = 2.0/1.25 = 1.60$$

$$K_t = 2.35$$

$$S = \pi d^3/32 = \pi (1.25)^3/32 = 0.192 \text{ IN}^3$$

$$\sigma_m = \frac{M_m}{S} = \frac{4F_m}{S} = \frac{4(500) \text{ LB}\cdot\text{IN}}{0.192 \text{ IN}^3} = 10430 \text{ PSI}$$

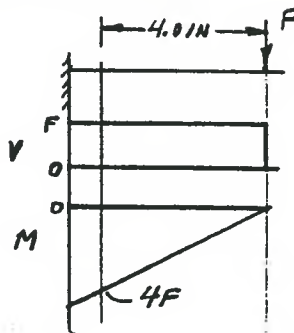
$$\sigma_a = \frac{M_a}{S} = \frac{4F_a}{S} = \frac{4(200) \text{ LB}\cdot\text{IN}}{0.192 \text{ IN}^3} = 4172 \text{ PSI}$$

$$\text{SAE 1050 HR: } S_y = 49 \text{ KSI}; S_u = 90 \text{ KSI}$$

$$S_m = 26 \text{ KSI (FIG 5-8 HR CURVE)}; S_m' = (0.95)(0.81)(26) = 20.0 \text{ KSI}$$

$$\frac{1}{N} = \frac{\sigma_m}{S_y} + \frac{K_t \sigma_a}{S_m'} = \frac{10430}{90000} + \frac{(2.35)(4172)}{20000} = 0.606; N = 1.65 \text{ LOW}$$

C_s - NON-ROTATING CIRC. SHAFT
 $D_e = 1.37 D = 0.463 \text{ IN}$
 $C_s = 0.95$



70. SEE PROB. 69: INCREASE N TO 3.0 OR HIGHER BY USING LARGER r . BEST POSSIBLE IMPROVEMENT WOULD BE $K_t = 1.0$ WITH LARGE r .

$$\frac{1}{N} = \frac{\sigma_m}{S_y} + \frac{K_t \sigma_a}{S_m'} = \frac{10430}{90000} + \frac{(1.0)(4172)}{20000} = 0.324; N = 3.08 \text{ OK}$$

CONSIDER A GRADUAL TAPER.

71. SEE PROB. 69: INCREASE MATERIAL STRENGTH TO GET $N \geq 3.0$.

TRY SAE 1340 OQT 900: $S_y = 158 \text{ KSI}; S_u = 169 \text{ KSI}$ (APP 4-3) 17% ELONG.

USE MACHINED SURFACE: $S_m = 58 \text{ KSI}; S_m' = (0.95)(0.81)(58) = 44.63 \text{ KSI}$

$$\frac{1}{N} = \frac{\sigma_m}{S_y} + \frac{K_t \sigma_a}{S_m'} = \frac{10430}{169000} + \frac{(2.35)(4172)}{44630} = 0.2814; N = 3.55 \text{ OK}$$

72. REPEATED REVERSED STRESS: CASE 4: DESIGN FOR $N \geq 3.0$.

SPECIFY MATERIAL: $\sigma_B = S_m'/N$.

$\sigma_{max, AT B}: M_B = 600 \text{ LB}\cdot\text{IN}$

$$S = \pi d^3/32 = \pi (1.00)^3/32 = 0.0982 \text{ IN}^3$$

$$\text{AT FILLET: } r/d = 0.06/1.0 = 0.06; D/d = 1.38/1.0 = 1.38$$

$$K_t = 1.99$$

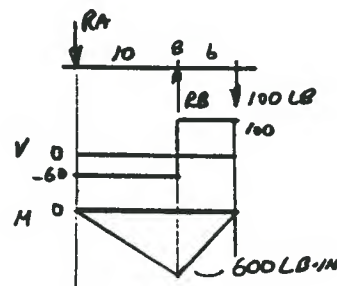
$$\text{LET } \sigma_a = \sigma_{max} = \frac{K_t M}{S} = \frac{1.99 (600) \text{ LB}\cdot\text{IN}}{0.0982 \text{ IN}^3} = 12159 \text{ PSI}$$

$$\text{REQ'D } S_m' = N \sigma' = (3.0)(12159) = 36477 \text{ PSI}$$

$$C_a = 0.81, C_s = 0.88; S_m = S_m'/(0.88)(0.81) = 51174 \text{ PSI}$$

FROM FIG. 5-8, REQ'D $S_m = 145000 \text{ PSI}$ (MACHINED SURFACE)

ONE POSSIBLE SOLUTION: SAE 3140 OQT 1000, $S_m = 152000 \text{ PSI}$, 17% ELONG.



73

FLUCTUATING NORMAL STRESS - CASE 5:

SAE 1340 OQT 700; $S_y = 197 \text{ ksi}$; $S_u = 221 \text{ ksi}$

$$S_u' = (0.80)(0.81)(65 \text{ ksi}) = 42.1 \text{ ksi}$$

AXIAL CR LFIG 5-8

$$F_m = (8500 + 16000)/2 = 12250 \text{ LB}$$

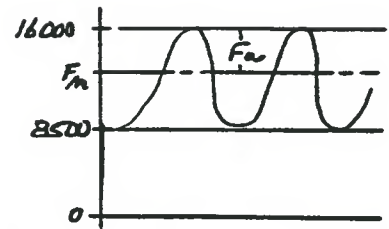
$$F_a = 16000 - 12250 = 3750 \text{ LB}$$

$$\text{AT FILLET: } r/d = 0.05/0.63 = 0.079; D/d = 1.00/0.63 = 1.59; K_t = 2.10$$

$$A = \pi d^2/4 = \pi (0.63)^2/4 = 0.312 \text{ in}^2$$

$$\sigma_m = F_m/A = \frac{12250 \text{ LB}}{0.312 \text{ in}^2} = 39300 \text{ psi}; \sigma_a = F_a/A = \frac{3750 \text{ LB}}{0.312 \text{ in}^2} = 12030 \text{ psi}$$

$$\frac{1}{N} = \frac{\sigma_m}{S_u} + \frac{K_t \sigma_a}{S_u'} = \frac{39300}{221000} + \frac{2.10(12030)}{42100} = 0.778; N = 1.29 \text{ LOW}$$



74

SEE PROB. 73: TRY TO REDESIGN TO ACHIEVE $N \geq 3.0$.INCREASE FILLET RADIUS TO $r = 0.185 \text{ in}$. FILLET WOULD THEN JUST BLEND WITH OUTSIDE OF 1.00 IN DIA.

$$r/d = \frac{0.185}{0.630} = 0.29; D/d = \frac{1.00}{0.63} = 1.59; K_t = 1.36$$

USE STRONGER MATERIAL: TRY SAE 8650 OQT 700

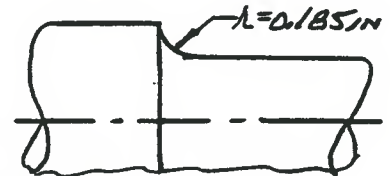
 $S_u = 240 \text{ ksi}$; $S_y = 222 \text{ ksi}$; 12% ELONGATIONGRIND ALL CRITICAL SURFACES GENTLY. $S_u = 88 \text{ ksi}$; USE $C_R = 0.81$

$$\text{THEN } S_u' = (0.81)(0.8)(88 \text{ ksi}) = 57.0 \text{ ksi}$$

$$\frac{1}{N} = \frac{39300}{240000} + \frac{(1.36)(12030)}{57000} = 0.451; N = 2.22 \text{ STILL TOO LOW}$$

EVEN IF $K_t = 1.0$, $N = 2.67$ - SLIGHTLY LOW

DIAMETERS MAY HAVE TO BE INCREASED.



75

REPEATED REVERSED LOAD - CASE 4:

BENDING MOMENT AT FILLETS $= F(4.00 \text{ in}) = (800)(4.0) = 3200 \text{ LB}\cdot\text{in}$.

$$S = \frac{M}{I} = \frac{(2.00)(1.25)^2}{6} = 0.5208 \text{ in}^3; \sigma_{nom} = \frac{M}{S} = \frac{3200 \text{ LB}\cdot\text{in}}{0.5208 \text{ in}^3} = 6144 \text{ psi}$$

$$N = \frac{S_u'}{\sigma_{max}} = \frac{S_u'}{K_t \sigma_{nom}}; \text{ THEN REQ'D } K_t = \frac{S_u'}{N \sigma_{nom}}$$

 C_s FOR RECTANGLE $1.25 \times 2.00 \text{ in}$. $D_e = 0.808 \sqrt{hb} = 0.808 \sqrt{(1.25)(2.00)} = 1.28$

$$C_s = (1.28/0.3)^{-0.11} = 0.85$$

AISI 1144 OQT 1100; $S_u = 112 \text{ ksi}$, $S_y = 42 \text{ ksi}$. USE $C_R = 0.81$

$$S_u' = C_s C_R S_u = (0.85)(0.81)(42) = 28.9 \text{ ksi} = 28900 \text{ psi}$$

$$\text{THEN } K_{t, \text{max}} = \frac{S_u'}{N \sigma_{nom}} = \frac{28900 \text{ psi}}{(3)(6144) \text{ psi}} = 1.57$$

FROM FATIGUE: $h/h_0 = 2.00/1.25 = 1.6$; FOR $K_t = 1.57$ THEN $r = 0.220 \text{ in}$ GIVES $K_t = 1.57$

76

DESIGN SECTION AT 'B' FOR $N \geq 3.0$.

$$\sum M_{PM} = 0 = 800(4.75) - R_1(2.25)$$

$$R_1 = 800(4.75)/2.25 = 1689 \text{ LB} \downarrow$$

MULTIPLE SOLUTIONS POSSIBLE:

LET WIDTH = 2.00 IN; SAME AS OTHER SECTIONS

DESIGN FOR $K_t \approx 1.50$

$$\text{CASE 4: } N = \frac{S_m'}{\sigma_{MAX}} = \frac{S_m'}{K_t \sigma_{NOM}}$$

$$\text{BUT } \sigma_{MAX} = \frac{M}{S} = \frac{M}{bh^2/6} = \frac{6M}{bh^2}$$

$$\text{THEN } N = \frac{S_m' bh^2}{K_t (6M)}$$

$$t = \sqrt{\frac{N K_t (6M)}{S_m' b}} = \left[\frac{3.0 \times 1.50 \times 6 \times (2533)}{(28900)(2.00)} \right]^{1/2} = 0.444 \text{ IN. MIN.}$$

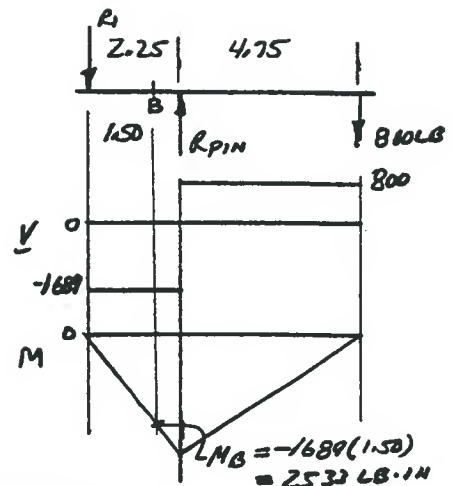
$$\text{LET } t = 0.85 \text{ IN} ; W = 2.00$$

$$: \text{ FOR } K_t = 1.50, N/t = 0.3 \text{ EST.}$$

$$\text{THEN } \lambda = 0.30 t = 0.30(0.85) = 0.255 \text{ IN 1ST. ESTIMATE.}$$

BY TRIAL

$$\text{FOR } \lambda = 0.19, K_t = 1.50$$



PROBLEMS 77-83 ARE DESIGN PROBLEMS FOR WHICH THERE ARE MULTIPLE SOLUTIONS POSSIBLE.

CHAPTER 6 COLUMNS

1. $r = D/4 = 0.75/4 = 0.188 \text{ in} ; KL/r = 1.0(32)/0.188 = 171$
 $S_y = 42000 \text{ psi} ; C_c = \sqrt{\frac{2\pi^2 E}{S_y}} = \sqrt{\frac{2\pi^2 (30 \times 10^9)}{42000}} = 119 \rightarrow \text{LONG COLUMN}$
 $A = \pi D^2/4 = 0.442 \text{ in}^2$
 $P_{CR} = \frac{\pi^2 EA}{(KL/r)^2} = \frac{\pi^2 (30 \times 10^6)(0.442)}{(171)^2} = 4473 \text{ LB}$
2. $KL/r = 1.0(15)/0.188 = 79.8 < C_c \rightarrow \text{SHORT-JOHNSON FORMULA}$
 $P_{CR} = AS_y \left[1 - \frac{S_y (KL/r)^2}{4\pi^2 E} \right] = (0.442)(42000) \left[1 - \frac{42000(79.8)^2}{4\pi^2 (30 \times 10^6)} \right] = 14373 \text{ LB}$
3. $r = 0.188 \text{ in} ; KL/r = 171 ; S_y = 21000 \text{ psi} ; E = 10 \times 10^6 \text{ psi}$
 $C_c = \sqrt{\frac{2\pi^2 (10 \times 10^6)}{21000}} = 97 \rightarrow \text{LONG COLUMN}$
 $P_{CR} = \frac{\pi^2 EA}{(KL/r)^2} = \frac{\pi^2 (10 \times 10^6)(0.442)}{(171)^2} = 1492 \text{ LB}$
4. $KL/r = (0.65)(32)/0.188 = 111 < C_c \rightarrow \text{SHORT-JOHNSON FORMULA}$
 $P_{CR} = (0.442)(42000) \left[1 - \frac{42000(111)^2}{4\pi^2 (30 \times 10^6)} \right] = 10500 \text{ LB}$
5. SQUARE: $r = S/\sqrt{12} = 0.65/\sqrt{12} = 0.188 \text{ in}$ SAME AS ROUND - PROB. 1.
 $P_{CR} = 4473 \text{ LB.}$
6. ACRYLIC: LET $S_y = \text{TENSILE STRENGTH} = 5400 \text{ psi} ; E = 220000 \text{ psi}$
 $C_c = \sqrt{\frac{2\pi^2 E}{S_y}} = \sqrt{\frac{2\pi^2 (220000)}{5400}} = 28.4 ; KL/r = 171 > C_c - \text{LONG}$
 $P_{CR} = \frac{\pi^2 EA}{(KL/r)^2} = \frac{\pi^2 (220000)(0.442)}{(171)^2} = 328 \text{ LB}$

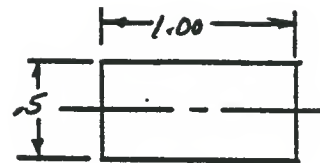
7.

$$r = 0.5 \text{ in} / \sqrt{12} = 0.144 \text{ in.}$$

$$KL/r = 1.0(8.5) / 0.144 = 58.9$$

$$S_y = 181000 \text{ psi} \rightarrow C_c = 57 \text{ (FIG. 6-4) LONG COL.}$$

$$P_{CR} = \frac{\pi^2 EA}{(KL/r)^2} = \frac{\pi^2 (30 \times 10^6) (0.5)}{(58.9)^2} = 42675 \text{ LB}$$



8.

$$r = \sqrt{(D^2 + d^2)/4} = \sqrt{(0.60^2 + 0.382^2)/4} = 0.529 \text{ in.}; L = 6.25 \text{ ft} \left(\frac{12 \text{ in}}{\text{ft}} \right) = 75 \text{ in}$$

$$A = \pi(D^2 - d^2)/4 = 0.5106 \text{ in}^2; L/r = 75/0.529 = 142$$

$$S_y = 30000 \text{ psi}; C_c = 140 \text{ (FIG. 6-5)}$$

$$a) \text{ PINNED ENDS: } KL/r = 1.0(L/r) = 142 > C_c \text{ LONG-EULER}$$

$$P_{CR} = \frac{\pi^2 (30 \times 10^6) (0.5106)}{(142)^2} = 7498 \text{ LB}$$

$$b) \text{ FIXED-FIXED: } KL/r = 0.65(L/r) = 92.3 < C_c \text{ - SHORT-JOHNSON}$$

$$P_{CR} = (0.5106) (30000) \left[1 - \frac{30000 (92.3)^2}{4 \pi^2 (30 \times 10^6)} \right] = 12000 \text{ LB}$$

$$c) \text{ FIXED-PINNED: } KL/r = 0.8(L/r) = 114 < C_c \text{ - SHORT-JOHNSON}$$

$$P_{CR} = (0.5106) (30000) \left[1 - \frac{30000 (114)^2}{4 \pi^2 (30 \times 10^6)} \right] = 10300 \text{ LB}$$

$$d) \text{ FIXED/FREE: } KL/r = 2.10(L/r) = 298 > C_c \text{ - LONG}$$

$$P_{CR} = \frac{\pi^2 (30 \times 10^6) (0.5106)}{(298)^2} = 1700 \text{ LB}$$

9.

$$S_y = 152 \text{ ksi}; C_c \approx 60 \text{ (FIG. 6-5)}; \text{ ASSUME COLUMN IS LONG: } K=1.0$$

$$(\text{EQ. 6-9}) D = \left[\frac{64NP(KL)^2}{\pi^3 E} \right]^{1/4} = \left[\frac{64(3)(8500)(50)^2}{\pi^3 (30 \times 10^6)} \right]^{1/4} = [4.39]^{1/4} = 1.45 \text{ in.}$$

$$r = D/4 = 1.50/4 = 0.375; KL/r = (1.0)(50)/0.375 = 133 > C_c \text{ LONG, OK.}$$

10.

$$S_y = 30 \text{ ksi}; C_c \approx 140 \text{ (FIG. 6-5)}; \text{ ASSUME COLUMN IS LONG}$$

$$D = 1.45 \text{ in (SAME AS PROB. 9)} \text{ USE } D = 1.50; KL/r = 133 < C_c \text{ - JOHNSON}$$

$$(\text{EQ. 6-10}) D = \left[\frac{4(3)(8500)}{\pi(30000)} + \frac{4(30000)(50)^2}{\pi^2 (30 \times 10^6)} \right]^{1/2} = 1.45 \text{ in USE } 1.50 \text{ in.}$$

NO ADVANTAGE TO 4140 STEEL

11.

$$\text{ALUM. 2014-T4: } S_y = 42000 \text{ psi}; C_c \approx 69; \text{ ASSUME LONG COLUMN}$$

$$D = \left[\frac{64(3)(8500)(50)^2}{\pi^2 (10 \times 10^6)} \right]^{1/4} = 1.90 \text{ in. - USED } 2.00 \text{ in, } r = D/4 = 0.50 \text{ in.}$$

$$KL/r = (1.0)(50)/0.50 = 100 > C_c \text{ LONG. D OK.}$$

12.

$$\text{SQUARE: } I = S^4/12; A = S^2; \text{ FROM EQ. 6-8 - EULER}$$

$$I = S^4/12 = NP(KL)^2 / \pi^2 E$$

$$S = \left[\frac{12NP(KL)^2}{\pi^2 E} \right]^{1/4}$$

(CONTINUE NEXT PAGE)

12. (CONTINUED) JOHNSON: $\lambda^2 = S^2/\pi^2$

$$\text{EQ. 6-7: } P_{CR} = NP = AS_y \left[1 - \frac{S_y (KL)^2}{4\pi^2 E \lambda^2} \right] = S^2 S_y \left[1 - \frac{S_y (KL)^2}{4\pi^2 E (S^2/\pi^2)} \right]$$

$$NP = S^2 S_y - \frac{S^2 S_y^2 (KL)^2}{4\pi^2 E S^2} = S^2 S_y - \frac{3S_y^2 (KL)^2}{\pi^2 E} \quad \text{SOLVE FOR } S$$

$$S^2 S_y = NP + \frac{3S_y^2 (KL)^2}{\pi^2 E} \quad \text{DIVIDE BY } S_y \text{ AND TAKE } \sqrt{}$$

$$S = \left[\frac{NP}{S_y} + \frac{3S_y (KL)^2}{\pi^2 E} \right]^{1/2}$$

13. EULER: EQ. (6-8) $I = NP(KL)^2/\pi^2 E$

$$I = \frac{\pi(D^4 - d^4)}{64} = \frac{\pi(D^4 - R^4 D^4)}{64} = \frac{\pi D^4 (1 - R^4)}{64}$$

$$\frac{\pi D^4 (1 - R^4)}{64} = \frac{NP(KL)^2}{\pi^2 E}$$

$$D = \left[\frac{64 NP (KL)^2}{\pi^3 E (1 - R^4)} \right]^{1/4}$$

JOHNSON: EQ. (6-7)

$$P_{CR} = NP = AS_y \left[1 - \frac{S_y (KL/\lambda)^2}{4\pi^2 E} \right]$$

$$\frac{NP}{S_y} = A - \frac{AS_y (KL)^2}{4\pi^2 E \lambda^2}$$

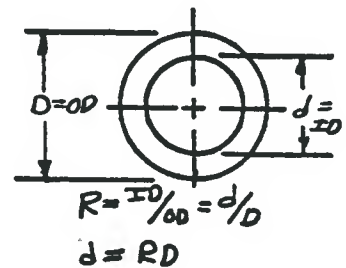
$$= \frac{\pi D^4 (1 - R^4)}{64} - \frac{\pi D^4 (1 - R^4) S_y (KL)^2}{(4\pi^2 E) \lambda^2 (1 - R^4) / 16}$$

$$\frac{NP}{S_y} = \frac{\pi D^4 (1 - R^4)}{64} - \frac{(1 - R^4) S_y (KL)^2}{\pi E (1 + R^2)}$$

$$\frac{\pi D^4 (1 - R^4)}{64} = \frac{NP}{S_y} + \frac{(1 - R^4) S_y (KL)^2}{\pi E (1 + R^2)}$$

$$D = \left[\frac{4NP}{\pi S_y (1 - R^4)} + \frac{4(1 - R^4) S_y (KL)^2}{\pi^2 E (1 + R^2) (1 - R^4)} \right]^{1/2}$$

$$D = \left[\frac{4NP}{\pi S_y (1 - R^4)} + \frac{4S_y (KL)^2}{\pi^2 E (1 + R^2)} \right]^{1/2}$$



$$A = \frac{\pi(D^2 - d^2)}{4} = \frac{\pi(D^2 - R^2 D^2)}{4}$$

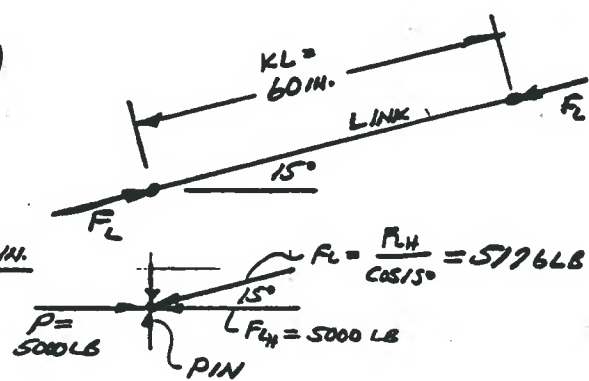
$$A = \pi D^2 (1 - R^2) / 4$$

$$\lambda^2 = \frac{D^2 + d^2}{16} = \frac{D^2 + R^2 D^2}{16}$$

$$\lambda^2 = \frac{D^2 (1 + R^2)}{16}$$

14. ASSUME COLUMN IS LONG: FROM PROB. 12: $S = \left[\frac{12NP(KL)^2}{\pi^2 E} \right]^{1/4}$
 $KL = 0.65(64) = 41.6 \text{ IN (FIXED ENDS)}$
 $S = \left[\frac{12(3)(6500)(41.6)^2}{\pi^2 (11 \times 10^6)} \right]^{1/4} = 1.423 \text{ IN} \rightarrow \text{USE } S = 1.500 \text{ IN}$
 CHECK: $\lambda = S/\sqrt{I_x} = 1.50/\sqrt{I_x} = 0.433$; $KL/\lambda = 41.6/0.433 = 96.1$
 FOR 6061-T6, $S_y = 40 \text{ ksi}$; FROM FIG (6-6) $C_c = 70$ LONG OK

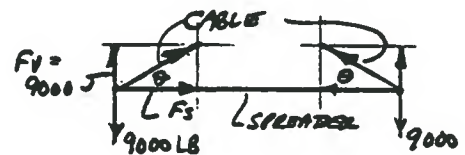
15. $R = ID/OD = 0.8$; $(1-R^4) = 0.5904$
 ASSUME LONG; FROM PROB. 13
 $D = \left[\frac{64NP(KL)^2}{4\pi^2 E (1-R^4)} \right]^{1/4} = \left[\frac{64(3)(6500)(41.6)^2}{4\pi^2 (10 \times 10^6)(0.5904)} \right]^{1/4} = 1.31 \text{ IN}$
 USE $D = 1.50 \text{ IN}$; $d = 0.8D = 1.20 \text{ IN}$; $r = \sqrt{\frac{D^2 + d^2}{4}} = 0.48 \text{ IN}$; $KL/r = 87$ LONG
 WEIGHT COMPARISON: WT. PROPORTIONAL TO AREA
 SQUARE: $A = S^2 = (1.50)^2 = 2.25 \text{ IN}^2$
 TUBE: $A = \frac{\pi}{4}(D^2 - d^2) = \frac{\pi}{4}(1.50^2 - 1.20^2) = 0.636 \text{ IN}^2$
 $N_S/N_T = 2.25/0.636 = 3.54$ — TUBE MUCH MORE EFFICIENT

16. ASSUME COLUMN IS LONG: EQ. (6-9)
 $D = \left[\frac{64NP(KL)^2}{\pi^3 E} \right]^{1/4}$
 $D = \left[\frac{64(3.5)(5176)(60)^2}{\pi^3 (30 \times 10^6)} \right]^{1/4} = 1.46 \text{ IN}$ USE $D = 1.50 \text{ IN}$
 CHECK $\lambda = D/4 = 0.375$
 $KL/\lambda = 60/0.375 = 160$ LONG OK
 $C_c \approx 60$ FIG. (6-5); $S_y = 157 \text{ ksi}$
 NOTE: CARE WOULD HAVE TO BE USED AT CONNECTIONS TO ENSURE AXIAL LOAD.
- 

17. MULTIPLE DESIGNS POSSIBLE. CONSIDER HOLLOW TUBE—ROUND OR SQUARE.
 CHEAPER MATERIAL MAY ALSO BE USED.

18. MULTIPLE DESIGNS POSSIBLE
 $F_s = F_v / \tan \theta = \frac{9000}{\tan 30^\circ} = 15588 \text{ LB}$

19. MULTIPLE DESIGNS POSSIBLE
 $F_s = \frac{9000}{\tan 15^\circ} = 33588 \text{ LB}$



20. $L = 10.75 \text{ FT} \times 12 \text{ in/ft} = 129 \text{ in.} ; KL = 2.1(129) = 271 \text{ in.} ; S_y = 68 \text{ ksi} ; C_c = 93$
 ASSUME COLUMN IS LONG-EULER - EQ. (6-9)

$$D = \left[\frac{64NP(KL)^2}{\pi^3 E} \right]^{1/4} = \left[\frac{64(25)(25000)(271)^2}{\pi^3 (30 \times 10^6)} \right]^{1/4} = 4.21 \text{ in.}$$

 USE $D = 4.25 \text{ in.}$
 CHECK $\lambda = D/4 = 1.063 \text{ in.} ; KL/\lambda = 271/1.063 = 255 - \text{LONG. OK}$

21. 22 MULTIPLE SOLUTIONS POSSIBLE

23. CROOKED COLUMN: $a = 0.08 \text{ in.} ; D = 0.75 \text{ in.} ; c = D/2 = 0.375 \text{ in.}$
 $\lambda = D/4 = 0.188 \text{ in.} ; \lambda^2 = 0.0352 \text{ in}^2 ; A = 0.442 \text{ in}^2 ; P_{cr} = 4473 \text{ LB}$
 IN EQ. 6-11: $C_1 = -\frac{1}{N} \left[S_y A + \left(1 + \frac{ac}{\lambda^2} \right) P_{cr} \right]$ [PROB. 6-1]

$$C_1 = -\frac{1}{3} \left[(42000)(0.442) + \left(1 + \frac{(0.08)(0.375)}{0.0352} \right) (4473) \right] = -8951$$

$$C_2 = \frac{S_y A P_{cr}}{N^2} = \frac{42000(0.442)(4473)}{(3)^2} = 9.226 \times 10^6$$

$$P = 0.5 \left[-(-8951) - \sqrt{(-8951)^2 - 4(9.226 \times 10^6)} \right] = 1189 \text{ LB}$$

24. CROOKED COLUMN: $a = 0.04 \text{ in.} ; c = 0.5/2 = 0.25 \text{ in.} ; P_{cr} = 42675 \text{ LB}$ (PROB. 6-9)
 $\lambda = 0.144 \text{ in.} ; \lambda^2 = 0.0208 \text{ in}^2 ; A = 0.50 \text{ in}^2 ; S_y = 181000 \text{ PSI}$

$$C_1 = -\frac{1}{3} \left[(181000)(0.50) + \left(1 + \frac{(0.04)(0.25)}{0.0208} \right) (42675) \right] = -51220$$

$$C_2 = \frac{(181000)(0.50)(42675)}{(3)^2} = 4.291 \times 10^8$$

$$P = 0.5 \left[-(-51220) - \sqrt{(-51220)^2 - 4(4.291 \times 10^8)} \right] = 10532 \text{ LB}$$

25. CROOKED COLUMN: $a = 0.15 \text{ in.} ; c = 0.8/2 = 0.40 \text{ in.} ; P_{cr} = 7498 \text{ LB}$ (PROB. 6-9)
 $\lambda = 0.529 \text{ in.} ; \lambda^2 = 0.280 \text{ in}^2 ; A = 0.5106 \text{ in}^2 ; S_y = 30000 \text{ PSI}$

$$C_1 = -\frac{1}{3} \left[(30000)(0.5106) + \left(1 + \frac{(0.15)(0.40)}{0.280} \right) (7498) \right] = -8677$$

$$C_2 = \frac{(30000)(0.5106)(7498)}{(3)^2} = 1.276 \times 10^7$$

$$P = 0.5 \left[-(-8677) - \sqrt{(-8677)^2 - 4(1.276 \times 10^7)} \right] = 1877 \text{ LB}$$

26

ECCENTRIC COLUMN: $L = 42 \text{ in.}$; $S = 1.25 \text{ in.}$; $C = \frac{S}{2} = 0.625 \text{ in.}$ $r = \frac{S}{\sqrt{12}} = 0.361 \text{ in.}$; $r^2 = 0.130 \text{ in}^2$; $A = S^2 = 1.563 \text{ in}^2$; $e = 0.60 \text{ in}$ $I = \frac{S^4}{12} = 0.2035 \text{ in}^4$; $S_y = 13000 \text{ psi}$ FOR AL. 6063-T4.

$$\text{FROM EQ. (6-12)} \quad \sigma_{\max} = \sigma_{\frac{1}{2}} = \frac{1250}{1.563} \left[1 + \frac{(0.60)(0.625)}{0.130} \sec \left(\frac{42}{2(0.361)} \sqrt{\frac{1250}{(1.563)(10 \times 10^6)}} \right) \right]$$

$$\sigma_{\max} = 3458 \text{ psi}$$

$$\text{EQ. (6-14)}: \quad \gamma_{\max} = 0.60 \left[\sec \left(\frac{42}{2(0.361)} \sqrt{\frac{1250}{(1.563)(10 \times 10^6)}} \right) - 1 \right] = 0.0915 \text{ in.}$$

27

ECCENTRIC COLUMN: $L = 3.2 \text{ m} = 3200 \text{ mm}$; $P = 30.5 \text{ kN} = 30500 \text{ N}$ 3-IN SCH. 40; $D_o = 3.50 \text{ in}$, $C = \frac{D_o}{2} = 1.75 \text{ in}$ (25.4 mm) $r = 1.16 \text{ in}$ (29.4 mm); $r^2 = 868 \text{ mm}^2$; $A = 2.23 \text{ in}^2$ (25.4^2) = 1439 mm^2 $I = 3.02 \text{ in}^4$ (25.4^4) = $1.251 \times 10^6 \text{ mm}^4$; $e = 150 \text{ mm}$ AISI 1020 HR; $E = 207 \text{ GPa} = 207 \times 10^9 \text{ Pa} = 207000 \text{ MPa} = 207000 \text{ N/mm}^2$

$$\text{EQ. (6-12)}: \quad \sigma_{\max} = \sigma_{\frac{1}{2}} = \frac{30500}{1439} \left[1 + \frac{(150)(1.75)}{868} \sec \left(\frac{3200}{2(29.46)} \sqrt{\frac{30500}{(1439)(207000)}} \right) \right]$$

 $\sigma_{\max} = 212 \text{ MPa}$; BUT $S_y = 207 \text{ MPa}$, THEN STRESS IS TOO HIGH.

$$\gamma_{\max} = (150) \left[\sec \left(\quad \right) - 1 \right] = 25.9 \text{ mm IF MATL. DOES NOT YIELD.}$$

28

ECCENTRIC COLUMN: $L = 14.75 \text{ in.}$; $C = 0.30 \text{ in.}$; $S = 0.250 \text{ in.}$ $A = S^2 = 0.0625 \text{ in}^2$; $r = \frac{S}{\sqrt{12}} = \frac{0.25}{\sqrt{12}} = 0.0722 \text{ in.}$; $r^2 = 0.00521 \text{ in}^2$ $P = 45 \text{ LB}$; $E = 28 \times 10^6 \text{ psi}$; $C = \frac{S}{2} = 0.125 \text{ in.}$

$$\text{EQ. (6-12)}: \quad \sigma_{\max} = \frac{45}{0.0625} \left[1 + \frac{(0.30)(0.125)}{0.00521} \sec \left(\frac{14.75}{2(0.0722)} \sqrt{\frac{45}{(0.0625)(28 \times 10^6)}} \right) \right] = 6685 \text{ psi}$$

$$\text{EQ. (6-14)}: \quad \gamma_{\max} = (0.30) \left[\sec \left(\quad \right) - 1 \right] = 0.045 \text{ in.}$$

29

ECCENTRIC COLUMN: $L = 40 \text{ in.}$; $C = 0.50 \text{ in.}$; $P = 75000 \text{ LB}$ FROM APP. 15-14: $A = 3.37 \text{ in}^2$; $r = 1.52 \text{ in.}$; $r^2 = 2.31 \text{ in}^2$; $C = \frac{L}{2} = 20 \text{ in.}$ ASTM A242: $S_y = 50000 \text{ psi}$; $E = 30 \times 10^6 \text{ psi}$ LET $N = 3$, THEN $\sigma_B = \frac{S_y}{3} = 16667 \text{ psi}$

$$\text{LET } \sigma_B' = \text{RIGHT SIDE OF EQ. (6-13)} = \frac{75000}{3.37} \left[1 + \frac{(0.50)(2.00)}{2.31} \sec \left(\frac{40}{2(1.52)} \sqrt{\frac{75000(3)}{(3.37)(30 \times 10^6)}} \right) \right]$$

$$\sigma_B' = 34099 \text{ psi} > \sigma_B \quad \text{UNSAFE / NOTE: } 6 \times 6 \times \frac{1}{2} \text{ REQ'D: } \sigma_B' = 9493 \text{ psi}$$

30

CENTRAL LOAD: $L = 16.0 \text{ FT}$ (12 in./ft) = 192 in. ASTM A36, $S_y = 36 \text{ ksi}$ FROM APP. 15-9: $A = 5.54 \text{ in}^2$; $I_y = 9.13 \text{ in}^4$; $r = \sqrt{I/A} = 1.28$; $KL = \frac{(0.8)(192)}{1.28} = 119.7$ FROM FIG. 6-5, $C_c = 125$; SHORT COLUMN - JOHNSON FORMULA. LET $N = 3$

$$P_a = \frac{P_{ce}}{N} = \frac{(5.54)(36000)}{3} \left[1 - \frac{(36000)(119.7)^2}{4\pi^2(30 \times 10^6)} \right] = 37500 \text{ LB}$$

31

CENTRAL LOAD: FIXED-END, $K=0.65$, $L_e = 0.85(66) = 42.9$ IN.

S4x7.7: $A = 2.26$ IN²; $r_{MN} = r_y = 0.581$ IN; $L_e/r = 73.8$; $N = 3$, $P_a = P_{cr}/N$

ASTM A36: $S_y = 36000$ PSI; $E = 30 \times 10^6$ PSI; $C_c \approx 130$ - SHORT COLUMN, JOHNSON EQ (6-7)

$$P_a = [(2.26)(36000)/3] \left[1 - \frac{(36000)(73.8)^2}{4\pi^2(30 \times 10^6)} \right] = \underline{22600 \text{ LB.}}$$

32

ECCENTRIC LOAD: $P = 1000$ LB; $e = 0.50 + 0.20/2 = 0.90$ IN; $L = 72$ IN.

$A = (1.60)(0.80) = 1.28$ IN²; $C = 0.80/2 = 0.40$ IN; $r = 0.80/\sqrt{12} = 0.2309$ IN.

USE STEEL - $E = 30 \times 10^6$ PSI

$$\sigma_{max} = \sigma_{1/2} = \frac{1000}{1.28} \left[1 + \frac{(0.90)(0.40)}{(0.2309)^2} \sec \left(\frac{72}{2(0.2309)} \sqrt{\frac{1000}{(1.28)(30 \times 10^6)}} \right) \right] = \underline{83150 \text{ PSI}}$$

$$M_{max} = 0.90 [\sec(1.7953) - 1] = \underline{0.386 \text{ IN.}}$$

SPECIFY A MATERIAL TO PROVIDE $N \geq 3$.

$$\text{USING EQ (6-13): } \sigma_a = \frac{1000}{1.28} \left[1 + \frac{(0.90)(0.40)}{(0.2309)^2} \sec \left(\frac{72}{2(0.2309)} \sqrt{\frac{3000}{AE}} \right) \right]$$

$$\sigma_a = 28280 \text{ PSI} = S_y/N; \text{ THEN } S_y = N \sigma_a = 84900 \text{ PSI}$$

SPECIFY AISI 1040 WQT 1000, $S_y = 86000$ PSI (OTHER SOLUTIONS POSSIBLE)

33

CENTRAL LOAD: SPECIFY A STEEL TUBE: $S_y = 36000$ PSI; LET $N = 3$

ASSUME COLUMN SUPPORTS $1/2$ TOTAL LOAD: $P_a = 55000 \text{ LB}/2 = 27500 \text{ LB}$

ASSUME COLUMN IS LONG: EQ. 6-8: $I = \frac{N P_a (KL)^2}{\pi^2 E}$ $C_c \approx 130$

ASSUME FIXED-PINNED COLUMN, $K = 0.8$: $KL = (0.8)(18.5 \text{ FT}) \frac{12 \text{ IN}}{\text{FT}} = 177.6$ IN. (LONG)

$$I = \frac{3(27500)(177.6)^2}{\pi^2(30 \times 10^6)} = 8.79 \text{ IN}^4; \text{ USE SQ. TUBE } 4 \times 4 \times 1/4, I = 11.9 \text{ IN}^4$$

OR RECT. TUBE $6 \times 4 \times 1/4$, $I_y = 11.1 \text{ IN}^4$ - LIGHTER

34

CENTRAL LOAD: C5x9 STEEL CHANNEL: $A = 2.64$ IN²; $r_y = 0.489$ IN; $N = 3$

$KL/r = 1.0(12)/0.489 = 229$: ASTM A36 - $S_y = 36000$ PSI, $C_c \approx 130$ - LONG (CL).

$$P_a = P_{cr}/N = \frac{\pi^2 EA}{N(KL/r)^2} = \frac{\pi^2(30 \times 10^6)(2.64)}{3(229)^2} = \underline{4967 \text{ LB}}$$

35

CENTRAL LOAD: SAME AS 34 EXCEPT FIXED ENDS, $K = 0.65$

$KL/r = (0.65)(12)/0.489 = 148.9$ - LONG COLUMN.

$$P_a = \frac{\pi^2(30 \times 10^6)(2.64)}{3(148.9)^2} = \underline{11750 \text{ LB}}$$

36

ECCENTRIC LOAD: $e = x$ FROM APP. 15 = 0.478 IN. AND $C = e$

USE EQ (6-13): $\sigma_a = S_y/N = 36000/3 = 12000$ PSI

$$\sigma_a = \frac{P_a}{2.64} \left[1 + \frac{(0.478)(0.478)}{(0.489)^2} \sec \left(\frac{112}{2(0.489)} \sqrt{\frac{3(P_a)}{(2.64)(30 \times 10^6)}} \right) \right]$$

BY ITERATION: FOR $P_a = 4100$ LB, $\sigma_a = 11920$ PSI

ECCENTRIC COLUMN ANALYSIS		Data from: Problem 6-37	
Solves Equation 6-13 for design stress and Equation 6-14 for maximum deflection			
Enter data for variables in <i>italics in shaded boxes</i>		Use consistent U.S. Customary units.	
Data To Be Entered:		Computed Values:	
Length and End Fixity:			
Column length, $L =$	126 in	Eq. Length, $L_e = KL =$ 126.0 in	
End fixity, $K =$	1		
Material Properties:			
Yield strength, $s_y =$	46000 psi	Column constant, $C_c =$ 111.6	
Modulus of Elasticity, $E =$	2.90E+07 psi	Argument for secant = 0.777 for strength Value of secant = 1.4025	
Cross Section Properties:		Argument for secant = 0.449 for deflection Value of secant = 1.1098	
[Note: Enter r or compute $r = \sqrt{I/A}$] [Always enter Area] [Enter zero for I or r if not used]			
Area, $A =$	6.020 in ²	Slenderness ratio, $KL/r =$ 89.4	
Moment of Inertia, $I =$	0 in ⁴		
Radius of Gyration, $r =$	1.410 in	Column is: short	
Values for Eqns. 6-13 and 6-14:		Req'd yield strength = 45,896 psi	
Eccentricity = $e =$	3 in	Must be less than actual yield strength:	
Neutral axis to outside = $c =$	2 in	$s_y =$ 46,000 psi	
Allowable load = $P_a =$	17600 lb	Max. deflection, $y_{max} =$ 0.329 in	
Design Factor			
Design factor on load, $N =$	3		

Note: A and r from Appendix 15-14

ECCENTRIC COLUMN ANALYSIS

Data from: Problem 38A US

Solves Equation 6-13 for design stress and Equation 6-14 for maximum deflection

Enter data for variables in italics in shaded boxes

Use consistent U.S. Customary units

Data To Be Entered:

Length and End Fixity:

Column length, $L = 40$ in

End fixity, $K = 1$

Material Properties:

Yield strength, $s_y = 40000$ psi

Modulus of Elasticity, $E = 1.00E+07$ psi

Cross Section Properties:

[Note: Enter r or compute $r = \sqrt{I/A}$]

[Always enter Area]

[Enter zero for I or r if not used]

Area, $A = 0.600$ in²

Moment of Inertia, $I = 0$ in⁴

OR

Radius of Gyration, $r = 0.433$ in

Values for Eqs. 6-13 and 6-14:

Eccentricity, $e = 1.75$ in

Neutral axis to outside, $c = 0.75$ in

Allowable load, $P_a = 685$ lb

Design Factor

Design factor on load, $N = 3$

Computed Values:

NOTE: This solution considers the eccentric load with bending about the horizontal axis.

Eq. Length, $L_e = KL = 40.0$ in

Column const., $C_c = 70.2$

Argument of sec = 0.855 for strength

Value of secant = 1.5236

Argument of sec = 0.494 for deflection

Value of secant = 1.1355

Slender. ratio, $KL/r = 92.4$

Column is: *long*

FINAL RESULTS

Req'd yield strength = 39,954 psi

Must be less than actual yield strength:

$s_y = 40,000$ psi

Max Deflection, $y_{max} = 0.237$ in

See also Solution 38B for buckling about the thinner vertical axis.

$$NOTE! A = (1.50 \text{ in})(0.40 \text{ in}) = 0.600 \text{ in}^2$$

$$r = \frac{H}{\sqrt{12}} = \frac{1.50 \text{ in}}{\sqrt{12}} = 0.433 \text{ in} ; P_a = 685 \text{ LB.}$$

ECCENTRIC LOAD TENDS TO BUCKLE THE BAR ABOUT ITS STRONG AXIS.
BUT SEE SOLUTION 38B. LIMITING LOAD IS 163 LB FOR BUCKLING ABOUT THIN AXIS.

COLUMN ANALYSIS PROGRAM		Data from: Problem 8-38B US
Refer to Figure 6-4 for analysis logic		
Enter data for variables in <i>italics in shaded boxes</i>		Use consistent U.S. Customary units.
Data To Be Entered:		Computed Values:
Length and End Fixity:		
Column length, <i>L</i> = 40 in		Eq. Length, <i>L_e</i> = <i>KL</i> = 40.0 in
End fixity, <i>K</i> = 1		
Material Properties:		
Yield strength, <i>s_y</i> = 40000 psi		Column const., <i>C_c</i> = 70.2
Modulus of Elasticity, <i>E</i> = 1.00E+07 psi		
Cross Section Properties:		NOTE: Cross section properties taken with respect to the vertical axis because the load is central to that axis. But buckling is expected about the axis through the thin (0.40 in) section.
[Note: Enter <i>r</i> or compute <i>r</i> = sqrt(<i>I/A</i>)]		
[Always enter Area]		
[Enter zero for <i>I</i> or <i>r</i> if not used]		
Area, <i>A</i> = 0.6 in ²		Slender. ratio, <i>KL/r</i> = 347.8
Moment of Inertia, <i>I</i> = 0 in ⁴		
Radius of Gyration, <i>r</i> = 0.115 in		Column is: long
Design Factor		Critical Buckling Load = 489 lb
Design factor on load, <i>N</i> = 3		Allowable Load = 163 lb
		This value governs the design, not solution 38A

ANALYSIS AS A STRAIGHT CENTRALLY LOADED COLUMN

THAT TENDS TO BUCKLE ABOUT THIN AXIS, *t* = 0.40 IN

$$r = \frac{t}{\sqrt{12}} = \frac{0.40 \text{ IN}}{\sqrt{12}} = 0.115 \text{ IN}$$

FROM EULER FORMULA WITH *N* = 3 *P_a* = 163 LB.

ECCENTRIC COLUMN ANALYSIS

Solves Equation 6-13 for design stress and Equation 6-14 for maximum deflection

Enter data for variables in italics in shaded boxes

Data from: Problem 39-SI

Use consistent SI Metric units

Data To Be Entered:

Length and End Fixity:

Column length, $L = 750 \text{ mm}$

End fixity, $K = 1$

Material Properties:

Yield strength, $s_y = 966 \text{ MPa}$

Modulus of Elasticity, $E = 200 \text{ GPa}$

Cross Section Properties:

[Note: Enter r or compute $r = \sqrt{I/A}$]

[Always enter Area]

[Enter zero for I or r if not used]

Area, $A = 314 \text{ mm}^2$

Moment of Inertia, $I = 0 \text{ mm}^4$

OR

Radius of Gyration, $r = 7.269 \text{ mm}$

Values for Eqns. 6-13 and 6-14:

Eccentricity, $e = 20 \text{ mm}$

Neutral axis to outside, $c = 12.5 \text{ mm}$

Allowable load, $P_a = 5200 \text{ N}$

Design Factor

Design factor on load, $N = 3$

Computed Values:

Eq. Length, $L_e = KL = 750.0 \text{ mm}$

Column const., $C_c = 63.9$

Argument of sec = 0.811 for strength

Value of secant = 1.4512

Argument of sec = 0.468 for deflection

Value of secant = 1.1205

Slender. ratio, $KL/r = 102.9$

Column is: *long*

FINAL RESULTS

Req'd yield strength = 389 MPa

Must be less than actual yield strength:

$s_y = 966 \text{ MPa}$

Max Deflection, $y_{\max} = 2.41 \text{ mm}$

Piston rod is safe for $P_a = 5200 \text{ N}$.

COLUMN ANALYSIS PROGRAM		Data from: Problem 40A-Straight
Refer to Figure 6-4 for analysis logic		
Enter data for variables in <i>italics</i> in shaded boxes		Use consistent U.S. Customary units.
Data To Be Entered:		Computed Values:
Length and End Fixity:		NOTE: Analysis of straight pipe. See also Solution 40B for crooked pipe.
Column length, $L = 156$ in		Eq. Length, $L_e = KL = 156.0$ in
End fixity, $K = 1$		
Material Properties:		
Yield strength, $s_y = 36000$ psi		Column const., $C_c = 128.3$
Modulus of Elasticity, $E = 3.00E+07$ psi		
Cross Section Properties:		
[Note: Enter r or compute $r = \sqrt{I/A}$]		
[Always enter Area]		
[Enter zero for I or r if not used]		
Area, $A = 1.075$ in ²		Slender. ratio, $KL/r = 198.2$
Moment of Inertia, $I = 0$ in ⁴		
or		Column is: <i>long</i>
Radius of Gyration, $r = 0.787$ in		Critical Buckling Load = 8,101 lb
		Straight Pipe
Design Factor		Allowable Load = 2,700 lb
Design factor on load, $N = 3$		See also Solution 40B for crooked pipe.

CROOKED COLUMN ANALYSIS

Solves Equation 6-11 for Allowable Load

Enter data for variables in *italics in shaded boxes*

Data To Be Entered:

Length and End Fixity:

Column length, $L = 156$ in

End fixity, $K = 1$

Material Properties:

Yield strength, $s_y = 36000$ psi

Modulus of Elasticity, $E = 3.00E+07$ psi

Cross Section Properties:

[Note: Enter r or compute $r = \sqrt{I/A}$]

[Always enter Area]

[Enter zero for I or r if not used]

Area, $A = 1.075$ in²

Moment of Inertia, $I = 0$ in⁴

Radius of Gyration, $r = 0.787$ in

Values for Eqn. 6-11:

Initial crookedness = $a = 1.25$ in

Neutral axis to outside = $c = 1.188$ in

Design Factor

Design factor on load, $N = 3$

Data from: Problem 40B-Crooked

Use consistent U.S. Customary units.

Computed Values:

Eq. Length, $L_e = KL = 156.0$ in

Column const., $C_c = 128.3$

Euler buckling load = 8101 lb

C_1 in Eqn. 6-11 = -22074

C_2 in Eqn. 6-11 = 3.483E+07

Slender. ratio, $KL/r = 198.2$

Column is: *long*

Straight Column

Critical Buckling Load = 8,101 lb

Crooked Column

Allowable Load = 1,711 lb

This value governs the use of the pipe.

See solution for straight pipe; Problem 40A.

CHAPTER 7

BELT DRIVES AND CHAIN DRIVES

V-BELTS

1. $C \leq 24.0 \text{ IN}$; $D_2 = 13.95 \text{ IN}$; $D_1 = 5.25 \text{ IN}$; 3V BELT
EQ. 7-31

$$L = 2(24) + 1.57(13.95 + 5.25) + \frac{(13.95 - 5.25)^2}{4(24)} = 78.73 \text{ IN}$$
 USE $L = 75 \text{ IN}$ - STANDARD LENGTH
2. ACTUAL C FROM EQ. 7-4: $L = 75$
 $B = 4(75) - 6.28(13.95 + 5.25) = 179.4$

$$C = \frac{179.4 + \sqrt{(179.4)^2 - 32(13.95 - 5.25)^2}}{16} = 22.00 \text{ IN}$$
3. $\theta_1 = 180^\circ - 2 \sin^{-1} \left[\frac{13.95 - 5.25}{2(22.0)} \right] = 157.2^\circ$ (EQ. 7-5)
 $\theta_2 = 180^\circ + 2 \sin^{-1} \left[\frac{13.95 - 5.25}{2(22.0)} \right] = 202.8^\circ$ (EQ. 7-6)
4. $C \leq 60.0 \text{ IN}$; $D_2 = 27.7 \text{ IN}$; $D_1 = 8.4 \text{ IN}$; 5V BELT (EQ. 7-3)
 COMPUTED $L = 178.2 \text{ IN}$. USE $L = 170 \text{ IN}$
5. ACTUAL $C = 55.83 \text{ IN}$. (EQ. 7-4)
6. $\theta_1 = 160.1^\circ$; $\theta_2 = 199.9^\circ$ (EQ. 7-5), (EQ. 7-6)
7. $C \leq 144 \text{ IN}$; $D_2 = 94.8 \text{ IN}$; $D_1 = 13.8 \text{ IN}$; 8V BELT (EQ. 7-3)
 COMPUTED $L = 469.9 \text{ IN}$. USE $L = 450 \text{ IN}$.
8. ACTUAL $C = 133.6 \text{ IN}$ (EQ. 7-4)
9. $\theta_1 = 144.7^\circ$; $\theta_2 = 215.3^\circ$ (EQS. 7-5, 7-6)
10. $N_b = R_1 \omega_1 = \frac{5.25 \text{ IN}}{2} \times \frac{1750 \text{ REV}}{\text{MIN}} \times \frac{2\pi \text{ RAD}}{\text{REV}} \times \frac{1 \text{ FT}}{12 \text{ IN}} = 2405 \text{ FT/MIN}$
11. $N_b = R_1 \omega_1 = \frac{8.4 \text{ IN}}{2} \times \frac{1160 \text{ REV}}{\text{MIN}} \times \frac{2\pi \text{ RAD}}{\text{REV}} \times \frac{1 \text{ FT}}{12 \text{ IN}} = 2551 \text{ FT/MIN}$
12. $N_b = R_1 \omega_1 = \frac{13.8 \text{ IN}}{2} \times \frac{870 \text{ REV}}{\text{MIN}} \times \frac{2\pi \text{ RAD}}{\text{REV}} \times \frac{1 \text{ FT}}{12 \text{ IN}} = 3143 \text{ FT/MIN}$
13. $\text{RATIO} = 13.95/5.25 = 2.66$; $P \approx 6.25 \text{ HP}$; $C_\theta = 0.94$; $C_L = 1.03$
 CORRECTED POWER = $C_\theta C_L P = (0.94)(1.03)(6.25) = 6.05 \text{ HP}$

V-BELTS

14. $RATIO = 27.7/8.4 = 3.30$; $P = 15.5 + 1.26 = 16.76 \text{ hp}$; $C_0 = .95$; $C_L = 1.05$
 CORRECTED POWER = $C_0 C_L P = (.95)(1.05)(16.76) = 16.72 \text{ hp}$
15. $RATIO = 94.8/13.8 = 6.87$; $P = 48 \text{ hp}$; $C_0 = .904$; $C_L = 1.09$
 CORRECTED POWER = $C_0 C_L P = (.904)(1.09)(48) = 47.3 \text{ hp}$
16. A 15N BELT IS A METRIC SIZE HAVING A TOP WIDTH OF 15MM SIMILAR TO A 5V BELT.
17. A 17A BELT IS A METRIC AUTOMOTIVE BELT HAVING A TOP WIDTH OF 17MM. SIMILAR TO A "3/4 IN." AUTOMOTIVE BELT
18. DESIGN: SERVICE FACTOR = 1.5; DESIGN POWER = $1.5(25) = 37.5 \text{ hp}$
 FROM FIG. 7-9 — USE 5V BELT
 $RATIO = 870/310 = 2.81$
 FOR $N_2 \approx 4000 \text{ FT/MIN}$; $D_1 = \frac{12(4000)}{\pi(870)} = 17.56 \text{ IN}$
 FOR $D_1 = 13.1 \text{ IN}$; $D_2 = 37.4 \text{ IN}$; $M_2 = 870 \times 13.1/37.4 = 305 \text{ RPM}$ OK
 RATED POWER = $22.5 + .94 = 23.44 \text{ hp}$
 CENTER DISTANCE:
 $D_2 < C < 3(D_2 + D_1)$
 $37.4 < C < 3(37.4 + 13.1) = 151.5$
 TRY $C \approx 48 \text{ IN}$. — NOMINAL $L = 178 \text{ IN}$ (EQ. 7-3)
 USE $L = 180 \text{ IN}$. — ACTUAL $C = 48.85 \text{ IN}$ (EQ. 7-4)
 $\theta_1 = 157.2^\circ$; $\theta_2 = 208.8^\circ$ (EQS. 7-5, 7-6)
 $C_0 = 0.92$; $C_L = 1.06$; CORR. POWER = $(.92)(1.06)(23.44) = 22.86 \text{ hp/BELT}$
 NO. OF BELTS = $37.5 \text{ hp} / 22.86 \text{ hp/BELT} = 1.64 \text{ BELTS} \rightarrow 2 \text{ BELTS}$
19. DESIGN: S.F. = 1.2; DES. POWER = $1.2(5) = 6.0 \text{ hp}$ — 3V BELT
 $RATIO = 1750/725 = 2.41$; $D_1 \approx 12(4000)/\pi(1750) = 8.7 \text{ IN}$
 FOR $D_1 = 7.95$; $D_2 = 18.95$; $M_2 = 1750 \times 7.95/18.95 = 734 \text{ RPM}$ OK
 RATED POWER $\approx 10.3 \text{ hp}$; $18.95 < C < 80.7$; TRY $C \approx 30 \text{ IN}$.
 $L \approx 103 \text{ IN}$ — USE $L = 100 \text{ IN}$; ACTUAL $C = 28.35 \text{ IN}$
 $\theta_1 = 157.6^\circ$; $\theta_2 = 202.4^\circ$; $C_0 = .94$; $C_L = 1.09$
 CORR. POWER = $(.94)(1.09)(10.3) = 10.55 \text{ hp/BELT}$ — ONE BELT REQ'D

V-BELTS

20.

DESIGN: $S.F. = 1.4$; DES. POWER = $1.4(40) = 56 \text{ HP} \rightarrow 5V \text{ BELT}$
 RATIO = $1500/550 = 2.73$; $D_1 \approx 12(4000)/\pi(1500) = 10.2 \text{ IN.}$
 FOR $D_1 = 10.2 \text{ IN.}$; $D_2 = 27.7 \text{ IN.}$; $M_2 = 1500 \times 10.2/27.7 = 552 \text{ RPM OK}$
 RATED POWER $\approx 25.7 \text{ HP}$ BY INTERPOLATION ON FIG 13-70 AT 1500 RPM
 $27.7 < C < 114$; USE $C \approx 36 \text{ IN.}$; $L \approx 133.6 \text{ IN.} \rightarrow \text{USE } L = 132 \text{ IN.}$
 ACTUAL $C = 35.16 \text{ IN.}$; $\theta_1 = 157.2^\circ$; $\theta_2 = 208.8^\circ$; $C_D = .92$; $C_L = 1.01$
 CORR. POWER = $(.92)(1.01)(25.7) = 23.9 \text{ HP/BELT}$
 NO. OF BELTS = $56/23.9 = 2.35 \rightarrow \text{USE 3 BELTS}$

21.

DESIGN: $S.F. = 1.4$; DES. POWER = $1.4(20) = 28 \text{ HP} \rightarrow 3V \text{ BELT}$
 RATIO = $1250/695 = 1.80$; $D_1 \approx 12(4000)/\pi(1250) = 12.2 \text{ IN.}$
 FOR $D_1 = 10.55 \text{ IN.}$; $D_2 = 18.95 \text{ IN.}$; $M_2 = 1250 \times 10.5/18.95 = 695.9 \text{ RPM OK}$
 RATED POWER $\approx 10.4 \text{ HP}$ BY INTERPOLATION ON FIG. 13-9 AT 1250 RPM
 $18.95 < C < 88.5$; USE $C \approx 20 \text{ IN.}$; $L \approx 87.2 \text{ IN.} \rightarrow \text{USE } L = 90 \text{ IN.}$
 ACTUAL $C = 21.43 \text{ IN.}$; $\theta_1 = 157.4^\circ$; $\theta_2 = 202.6^\circ$; $C_D = .94$; $C_L = 1.07$
 CORR. POWER = $(.94)(1.07)(10.4) = 10.46 \text{ HP/BELT}$
 NO. OF BELTS = $28/10.46 = 2.68 \text{ BELTS} \rightarrow \text{USE 3 BELTS}$

22.

DESIGN: $S.F. = 2.0$ (CHOKING); DES. P = $2.0(100) = 200 \text{ HP} \rightarrow 5V \text{ BELT}$
 RATIO = $870/625 = 1.39$; $D_1 \approx 12(4000)/\pi(870) = 17.6 \text{ IN.}$
 FOR $D_1 = 10.8 \text{ IN.}$; $D_2 = 14.9 \text{ IN.}$; $M_2 = 870 \times 10.8/14.9 = 631 \text{ RPM OK}$
 RATED POWER = $17.6 + .77 = 18.37 \text{ HP}$; $14.9 < C < 77.1$; USE $C \approx 48 \text{ IN.}$
 $L \approx 136 \text{ IN.} \rightarrow \text{USE } L = 132 \text{ IN.}$; ACTUAL $C = 45.78 \text{ IN.}$
 $\theta_1 = 174.9^\circ$; $\theta_2 = 185.1^\circ$; $C_D = .988$; $C_L = 1.01$
 CORR. POWER = $(.988)(1.01)(18.37) = 18.33 \text{ HP/BELT}$
 NO. OF BELTS = $200/18.33 = 10.9 \rightarrow \text{USE 11 BELTS, NOT ACCEPTABLE}$

TRY 8V BELT: FOR $D_1 = 17.8 \text{ IN.}$; $D_2 = 24.8 \text{ IN.}$; $M_2 = 624.4 \text{ RPM OK}$
 RATED POWER = 66 HP ; $24.8 < C < 127.8$; USE $C \approx 48 \text{ IN.}$
 $L \approx 163 \text{ IN.} \rightarrow \text{USE } L = 160 \text{ IN.}$; ACTUAL $C = 46.43 \text{ IN.}$
 $\theta_1 = 171.4^\circ$; $\theta_2 = 188.6^\circ$; $C_D = .98$; $C_L = .94$
 CORR. POWER = $(.98)(.94)(66) = 60.8 \text{ HP/BELT}$
 NO. OF BELTS = $200/60.8 = 3.3 \rightarrow \text{USE 4 BELTS}$

ROLLER CHAIN

23. CHAIN NO. 140: $PITCH = 1\frac{1}{8} = 1\frac{3}{4} \text{ IN.}$
24. CHAIN NO. 60: $PITCH = \frac{1}{8} = \frac{3}{4} \text{ IN.}$
25. STATIC LOAD = 1250 LB; AVG. TENSILE STRENGTH = $10W = 12500 \text{ LB.}$
USE A NO. 80 CHAIN (1.00 IN. PITCH); T.S. = 14500, TABLE 7-5
26. LOAD ON EACH CHAIN = 2500 LB; T.S. = $10W = 25000 \text{ LB}$
USE NO. 120 CHAIN $1\frac{1}{2} \text{ IN. PITCH}$; T.S. = 34000 LB
27. FATIGUE OF LINK PLATES; IMPACT OF ROLLERS ON SPROCKET TEETH; GALLING BETWEEN PINS AND BUSHINGS
28. TABLE 7-8: GIVEN NO. 60 CHAIN, 20 TEETH, 750 RPM
RATED POWER = 21.69 HP (INTERPOLATION), TYPE B LUBE (BATH)
SERVICE FACTOR = 1.2 FOR HYDRAULIC DRIVE
DESIGN POWER RATING = $21.69 / 1.2 = 18.08 \text{ HP.}$
29. 3 STRANDS: FACTOR = 2.5
POWER RATING = $2.5(18.08) = 45.2 \text{ HP}$
30. TABLE 7-7: GIVEN NO. 40 CHAIN, 12 TEETH, 860 RPM
RATED POWER = 4.44 HP (INTERPOLATION), TYPE B LUBE (BATH)
S.F. = 1.2; DESIGN POWER RATING = $4.44 / 1.2 = 3.70 \text{ HP.}$
31. 4 STRANDS: FACTOR = 3.3; POWER RATING = $3.3(3.70) = 12.21 \text{ HP}$
32. TABLE 7-9: GIVEN NO. 80 CHAIN; 32 TEETH; 1160 RPM
RATED POWER = 78.69 HP (INTERPOLATION), TYPE C LUBE (OIL STREAM)
SF = 1.2; DESIGN POWER = $78.69 / 1.2 = 65.57 \text{ HP}$
33. 2 STRANDS: FACTOR = 1.7; POWER RATING = $(1.7)65.57 \text{ HP} = \underline{111.5 \text{ HP}}$
34. NO. 60 CHAIN; $N_1 = 15$; $N_2 = 50$; $C \leq 36 \text{ IN.}$ — USE EQ. 7-9
 $L_P = \frac{3}{4} \text{ IN.} = 0.75 \text{ IN.}$
 $C = 36 \text{ IN.} / 0.75 \text{ IN.} / \text{PITCH} = 48 \text{ PITCHES}$
 $L = 2(48) + \frac{50+15}{2} + \frac{(50-15)^2}{4 \pi^2 (48)} = 129.1 \text{ PITCHES}$
USE 128 PITCHES (EVEN NUMBER); $L = 128(0.75) = 96 \text{ IN.}$

35. FOR $L=128$ PITCHES; C FROM EQ. 7-10

$$C = \frac{1}{4} \left[128 - \frac{50+15}{2} + \sqrt{\left[128 - \frac{50+15}{2} \right]^2 - \frac{8(50-15)^2}{4\pi^2}} \right] = 47.42 \text{ PITCHES}$$

$C = 47.42 \text{ PITCHES} \times 0.75 \text{ IN/PITCH} = 35.57 \text{ IN.}$

36. NO. 40 CHAIN; $N_1=11$; $N_2=45$; $C \leq 24 \text{ IN.}$
 $L/P = 1/2 \text{ IN.} = 0.50 \text{ IN.}; C = 24/0.5 = 48 \text{ PITCHES}$

$$L = 2(48) + \frac{45+11}{2} + \frac{(45-11)^2}{4\pi^2(48)} = 124.6 \text{ PITCHES} \rightarrow \text{USE } 124 \text{ PITCHES}$$

$\frac{124(0.5)}{124(0.5)} = 62 \text{ IN.}$

37. $C = \frac{1}{4} \left[124 - \frac{45+11}{2} + \sqrt{\left[124 - \frac{45+11}{2} \right]^2 - \frac{8(45-11)^2}{4\pi^2}} \right] = 47.69 \text{ PITCHES}$

$C = 47.69 \text{ PITCHES} \times 0.5 \text{ IN/PITCH} = 23.85 \text{ IN.}$

38. DESIGN: 25 HP; $M_1=310 \text{ RPM}$; $M_2=160 \text{ RPM}$; NOM. RATIO $= 310/160 = 1.94$
 $SF=1.5$; DESIGN POWER $= 1.5(25) = 37.5 \text{ HP}$
 USE 3 STRANDS; RATING $= 37.5/2.5 = 15.0 \text{ HP PER STRAND}$
 NO. 80 CHAIN; 15 TEETH RATED $> 15.0 \text{ HP AT } 310 \text{ RPM}$; TYPE B LUBE
 $N_2 = N_1 \times \text{RATIO} = 15(1.94) = 29.1 \rightarrow 29 \text{ TEETH}$
 $M_2 = M_1 \times N_1/N_2 = 310 \times 15/29 = 160.3 \text{ RPM OK}$
 $D_1 = 1.00/\sin(180/15) = 4.810 \text{ IN.}; D_2 = 1.00/\sin(180/29) = 9.249 \text{ IN.}$
 USE $C \approx 40$ PITCHES WITH $P=1.00 \text{ IN}$ FOR NO. 80 CHAIN

$$L = 2(40) + \frac{29+15}{2} + \frac{(29-15)^2}{4\pi^2(40)} = 102.1 \text{ PITCHES} \rightarrow \text{USE } 102 \text{ PITCHES} = L$$

$$C = \frac{1}{4} \left[102 - \frac{29+15}{2} + \sqrt{\left[102 - \frac{29+15}{2} \right]^2 - \frac{8(29-15)^2}{4\pi^2}} \right] = 39.94 \text{ PITCHES} = C$$

$C = 39.94 \text{ IN.}$

Problems 38-42 are design problems for chain drives for which there are no unique solutions. The general procedure is illustrated above for one possible solution for Problem 38. This and the other design problems are shown on the following pages using the spreadsheet that is available from the publisher's website for this book. Data for design power per strand from Tables 7-7, 7-8, or 7-9 must be used to ensure that the selected chain design has sufficient capacity.

CHAIN DRIVE DESIGN

Initial Input Data:

Problem 38 - Multiple strands

Application: Hammer Mill
Drive type: Electric motor
Driven machine: Hammer Mill
Power input: 25 hp
Service factor: 1.5
Input speed: 310 rpm
Desired output speed: 160 rpm

Table 7-1

Computed Data:

Design power: 37.5 hp
Speed ratio: 1.94

Design Decisions-Chain Type and Teeth Numbers:

Number of strands	3	1	2	3	4
Strand factor	2.5	1.0	1.7	2.5	3.3

Required power per strand: 15.00 hp

Chain number: 80 Tables 7-7, 7-8 or 7-9

Chain pitch: 1 in

Number of teeth-Driver sprocket: 15

Computed no. of teeth-Driver sprocket: 29.06

Enter Chosen number of teeth: 29

Computed Data:

Actual output speed: 160.3 rpm

Pitch diameter-Driver sprocket: 4.810 in

Pitch diameter-Driven sprocket: 9.249 in

Center Distance, Chain Length and Angle of Wrap:

Enter Nominal center distance: 40 pitches 30 to 50 pitches recommended

Computed nominal chain length: 102.1 pitches

Enter Specified no. of pitches: 102 pitches Even number recommended

Actual chain length: 102.00 in

Computed actual center distance: 39.938 pitches

Actual center distance: 39.938 in

Angle of wrap-Driver sprocket: 173.6 degrees Should be greater than 120 degrees

Angle of wrap-Driven sprocket: 186.4 degrees

CHAIN DRIVE DESIGN

Initial Input Data: *Problem 39 - Single strand*

Application: *Agitator*
Drive type: *Electric motor*
Driven machine: *Agitator*
Power input: *5 hp*
Service factor: *1*
Input speed: *750 rpm*
Desired output speed: *325 rpm*

Table 7-1

Computed Data:

Design power: *5 hp*
Speed ratio: *2.31*

Design Decisions-Chain Type and Teeth Numbers:

Number of strands:	<i>1</i>	<i>1</i>	<i>2</i>	<i>3</i>	<i>4</i>
Strand factor:	<i>1.0</i>	<i>1.0</i>	<i>1.7</i>	<i>2.5</i>	<i>3.3</i>

Required power per strand: *5.00 hp*

Chain number: *40* Tables 7-1, 7-8, or 7-9

Chain pitch: *0.5 in*

Number of teeth-Driver sprocket: *18*

Computed no. of teeth-Driver sprocket: *36.92*

Enter: Chosen number of teeth: *37*

Computed Data:

Actual output speed: *324.3 rpm*

Pitch diameter-Driver sprocket: *2.563 in*

Pitch diameter-Driven sprocket: *5.896 in*

Center Distance, Chain Length and Angle of Wrap:

Enter: Nominal center distance: *32 pitches* 30 to 50 pitches recommended

Computed nominal chain length: *90.8 pitches*

Enter: Specified no. of pitches: *90 pitches* Even number recommended

Actual chain length: *45.00 in*

Computed actual center distance: *31.573 pitches*

Actual center distance: *15.787 in*

Angle of wrap-Driver sprocket: *167.9 degrees* Should be greater than 120 degrees

Angle of wrap-Driven sprocket: *192.1 degrees*

CHAIN DRIVE DESIGN

Initial Input Data:

Problem 40 - Multiple strands

Application: Conveyor
 Drive type: Engine
 Driven machine: Heavy conveyor
 Power input: 40 hp
 Service factor: 1.4
 Input speed: 500 rpm
 Desired output speed: 250 rpm

Table 7-1

Computed Data:

Design power: 56 hp
 Speed ratio: 2.00

Design Decisions-Chain Type and Teeth Numbers:

Number of strands	3	1	2	3	4
Strand factor	2.5	1.0	1.7	2.5	3.3

Required power per strand: 22.40 hp

Chain number: 80 Tables 7-7, 7-8 or 7-9

Chain pitch: 1.00 in

Number of teeth-Driver sprocket: 14

Computed no. of teeth-Driver sprocket: 28.00

Enter: Chosen number of teeth: 28

Computed Data:

Actual output speed: 250.0 rpm
 Pitch diameter-Driver sprocket: 4.494 in
 Pitch diameter-Driven sprocket: 8.931 in

Center Distance, Chain Length and Angle of Wrap:

Enter: Nominal center distance: 36 pitches 30 to 50 pitches recommended

Computed nominal chain length: 93.1 pitches

Enter: Specified no. of pitches: 94 pitches Even number recommended

Actual chain length: 94.00 in

Computed actual center distance: 36.432 pitches

Actual center distance: 36.432 in

Angle of wrap-Driver sprocket: 173.0 degrees Should be greater than 120 degrees

Angle of wrap-Driven sprocket: 187.0 degrees

CHAIN DRIVE DESIGN

Initial Input Data:

Problem 41 - Multiple strands

Application: Pump drive
 Drive type: Steam turbine
 Driven machine: Centrifugal pump
 Power input: 20 hp
 Service factor: 1
 Input speed: 2200 rpm
 Desired output speed: 775 rpm

Table 7-1

Computed Data:

Design power: 20 hp
 Speed ratio: 2.84

Design Decisions-Chain Type and Teeth Numbers:

Number of strands	2	1	2	3	4
Strand factor	1.7	1.0	1.7	2.5	3.3

Required power per strand: 11.76 hp

Chain number: 40 Tables 7-7, 7-8 or 7-9

Chain pitch: 0.50 in

Number of teeth-Driver sprocket: 25 11.95 hp rating at 2200 rpm

Computed no. of teeth-Driver sprocket: 70.97

Enter Chosen number of teeth: 71 Check availability from vendor

Computed Data:

Actual output speed: 774.6 rpm

Pitch diameter-Driver sprocket: 3.989 in

Pitch diameter-Driven sprocket: 11.304 in

Center Distance, Chain Length and Angle of Wrap:

Enter Nominal center distance: 30 pitches 30 to 50 pitches recommended

Computed nominal chain length: 109.8 pitches

Enter Specified no. of pitches: 110 pitches Even number recommended

Actual chain length: 55.00 in

Computed actual center distance: 30.110 pitches

Actual center distance: 15.055 in

Angle of wrap-Driver sprocket: 151.9 degrees Should be greater than 120 degrees

Angle of wrap-Driven sprocket: 208.1 degrees

CHAIN DRIVE DESIGN**Initial Input Data:****Problem 42 - Multiple strands**

Application: Rock Crusher
Drive type: Hydraulic drive
Driven machine: Rock Crusher
Power input: 100 hp
Service factor: 1.4
Input speed: 625 rpm
Desired output speed: 225 rpm

Table 7-1

Computed Data:

Design power: 140 hp
Speed ratio: 2.78

Design Decisions-Chain Type and Teeth Numbers:

Number of strands:	4	1	2	3	4
Strand factor:	3.3	1.0	1.7	2.5	3.3

Required power per strand: 42.42 hp

Chain number: 80 Tables 7-7, 7-8 or 7-9

Chain pitch: 1.00 in

Number of teeth-Driver sprocket: 21 >42.94 hp per strand

Computed no. of teeth-Driver sprocket: 58.33

Enter, Chosen number of teeth: 58 Check availability from vendor

Computed Data:

Actual output speed: 226.3 rpm
Pitch diameter-Driver sprocket: 6.710 in
Pitch diameter-Driven sprocket: 18.471 in

Center Distance, Chain Length and Angle of Wrap:

Enter, Nominal center distance: 40 pitches 30 to 50 pitches recommended

Computed nominal chain length: 120.4 pitches

Enter, Specified no. of pitches: 120 pitches Even number recommended

Actual chain length: 120.00 in

Computed actual center distance: 39.815 pitches

Actual center distance: 39.815 in

Angle of wrap-Driver sprocket: 163.0 degrees Should be greater than 120 degrees

Angle of wrap-Driven sprocket: 197.0 degrees

CHAPTER 8 KINEMATICS OF GEARS

Gear Geometry

1. $N = 44; P_d = 12$

<p>a. $D = N/P_d = 44/12 = 3.667 \text{ in.}$</p> <p>b. $P_c = \pi/P_d = \pi/12 = 0.2618 \text{ in.}$</p> <p>c. $m = 25.4/P_d = 25.4/12 = 2.117 \text{ mm}$</p> <p>d. $m = 2.00 \text{ mm}$</p> <p>e. $a = 1/P_d = 1/12 = 0.0833 \text{ in.}$</p>	<p>f. $b = 1.25/P_d = 1.25/12 = 0.1042 \text{ in.}$</p> <p>g. $c = 0.25/P_d = 0.25/12 = 0.0208 \text{ in.}$</p> <p>h. $h_t = a + b = 2.25/P_d = 0.1875 \text{ in.}$</p> <p>i. $h_k = 2a = 2/P_d = 2/12 = 0.1667 \text{ in.}$</p> <p>j. $t = \pi/2P_d = \pi/2(12) = 0.131 \text{ in.}$</p> <p>k. $D_o = (N+2)/P_d = 46/12 = 3.833 \text{ in.}$</p>
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2. $N = 34; P_d = 24$

<p>a. $D = 34/24 = 1.417 \text{ in.}$</p> <p>b. $P_c = \pi/24 = 0.131 \text{ in.}$</p> <p>c. $m = 25.4/24 = 1.058 \text{ mm}$</p> <p>d. $m = 1.00 \text{ mm}$</p> <p>e. $a = 1/24 = 0.0417 \text{ in.}$</p>	<p>f. $b = \frac{1.200}{P_d} + 0.002 = 0.0520 \text{ in.}$</p> <p>g. $c = 0.200/P_d + 0.002 = 0.0103 \text{ in.}$</p> <p>h. $h_t = a + b = 0.0417 + 0.0520 = 0.0937 \text{ in.}$</p> <p>i. $h_k = 2a = 2/24 = 0.0833 \text{ in.}$</p> <p>j. $t = \pi/2(24) = 0.0654 \text{ in.}$</p> <p>k. $D_o = (N+2)/P_d = 36/24 = 1.500 \text{ in.}$</p>
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3. $N = 45; P_d = 2$

<p>a. $D = 45/2 = 22.500 \text{ in.}$</p> <p>b. $P_c = \pi/2 = 1.571 \text{ in.}$</p> <p>c. $m = 25.4/2 = 12.7 \text{ mm}$</p> <p>d. $m = 12 \text{ mm}$</p> <p>e. $a = 1/2 = 0.500 \text{ in.}$</p>	<p>f. $b = 1.25/2 = 0.625 \text{ in.}$</p> <p>g. $c = 0.25/2 = 0.125 \text{ in.}$</p> <p>h. $h_t = 2.25/2 = 1.125 \text{ in.}$</p> <p>i. $h_k = 2/2 = 1.000 \text{ in.}$</p> <p>j. $t = \pi/2(2) = 0.7854 \text{ in.}$</p> <p>k. $D_o = 47/2 = 23.50 \text{ in.}$</p>
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4. $N = 18; P_d = 8$

<p>a. $D = 18/8 = 2.250 \text{ in.}$</p> <p>b. $P_c = \pi/8 = 0.3927 \text{ in.}$</p> <p>c. $m = 25.4/8 = 3.175 \text{ mm}$</p> <p>d. $m = 3.0 \text{ mm}$</p> <p>e. $a = 1/8 = 0.125 \text{ in.}$</p>	<p>f. $b = 1.25/8 = 0.1563 \text{ in.}$</p> <p>g. $c = 0.25/8 = 0.0313 \text{ in.}$</p> <p>h. $h_t = 2.25/8 = 0.2813 \text{ in.}$</p> <p>i. $h_k = 2/8 = 0.250 \text{ in.}$</p> <p>j. $t = \pi/2(8) = 0.1963 \text{ in.}$</p> <p>k. $D_o = 20/8 = 2.500 \text{ in.}$</p>
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5. $N = 22; P_d = 1.75$

<p>a. $D = 22/1.75 = 12.571 \text{ in.}$</p> <p>b. $P_c = \pi/1.75 = 1.795 \text{ in.}$</p> <p>c. $m = 25.4/1.75 = 14.514 \text{ mm}$</p> <p>d. $m = 16 \text{ mm}$</p> <p>e. $a = 1/1.75 = 0.5714 \text{ in.}$</p>	<p>f. $b = 1.25/1.75 = 0.7143 \text{ in.}$</p> <p>g. $c = 0.25/1.75 = 0.1429 \text{ in.}$</p> <p>h. $h_t = 2.25/1.75 = 1.2857 \text{ in.}$</p> <p>i. $h_k = 2/1.75 = 1.1429 \text{ in.}$</p> <p>j. $t = \pi/2(1.75) = 0.8976 \text{ in.}$</p> <p>k. $D_o = 24/1.75 = 13.714 \text{ in.}$</p>
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6. $N=20; P_d=64$

a. $D = 20/64 = 0.3125 \text{ in.}$
 b. $P_c = \pi/64 = 0.0491 \text{ in.}$
 c. $m = 25.4/64 = 0.397 \text{ mm}$
 d. $m = 0.40 \text{ mm}$
 e. $a = 1/64 = 0.0156 \text{ in.}$

f. $b = 1.200/64 + 0.002 = 0.0208 \text{ in.}$
 g. $c = 0.200/64 + 0.002 = 0.0051 \text{ in.}$
 h. $h_t = a + b = 0.0364 \text{ in.}$
 i. $h_k = 2/64 = 0.0313 \text{ in.}$
 j. $t = \pi/2(64) = 0.0245 \text{ in.}$
 k. $D_o = 22/64 = 0.3438 \text{ in.}$

7. $N=180; P_d=80$

a. $D = 180/80 = 2.250 \text{ in.}$
 b. $P_c = \pi/80 = 0.0393 \text{ in.}$
 c. $m = 25.4/80 = 0.318 \text{ mm}$
 d. $m = 0.30 \text{ mm}$
 e. $a = 1/80 = 0.0125 \text{ in.}$

f. $b = 1.200/80 + 0.002 = 0.0170 \text{ in.}$
 g. $c = 0.200/80 + 0.002 = 0.0045 \text{ in.}$
 h. $h_t = a + b = 0.0295 \text{ in.}$
 i. $h_k = 2/80 = 0.025$
 j. $t = \pi/2(80) = 0.0196 \text{ in.}$
 k. $D_o = 182/80 = 2.275 \text{ in.}$

8. $N=28; P_d=18$

a. $D = 28/18 = 1.556 \text{ in.}$
 b. $P_c = \pi/18 = 0.1745 \text{ in.}$
 c. $m = 25.4/18 = 1.411 \text{ mm}$
 d. $m = 1.5 \text{ mm}$
 e. $a = 1/18 = 0.0556 \text{ in.}$

f. $b = 1.25/18 = 0.0694 \text{ in.}$
 g. $c = 0.25/18 = 0.0139 \text{ in.}$
 h. $h_t = 2.25/18 = 0.125 \text{ in.}$
 i. $h_k = 2/18 = 0.1111 \text{ in.}$
 j. $t = \pi/2(18) = 0.0873 \text{ in.}$
 k. $D_o = 30/18 = 1.667 \text{ in.}$

9. $N=28; P_d=20$

a. $D = 28/20 = 1.400 \text{ in.}$
 b. $P_c = \pi/20 = 0.1571 \text{ in.}$
 c. $m = 25.4/20 = 1.27 \text{ mm}$
 d. $m = 1.25 \text{ mm}$
 e. $a = 1/20 = 0.050 \text{ in.}$

f. $b = 1.200/20 + 0.002 = 0.0620 \text{ in.}$
 g. $c = 0.200/20 + 0.002 = 0.012 \text{ in.}$
 h. $h_t = a + b = 0.1120 \text{ in.}$
 i. $h_k = 2/20 = 0.1000 \text{ in.}$
 j. $t = \pi/2(20) = 0.0785 \text{ in.}$
 k. $D_o = 30/20 = 1.500 \text{ in.}$

10. $N=34; m=3=D/N$

a. $D = mN = 3(34) = 102 \text{ mm}$
 b. $P_c = \pi D/N = \pi m = \pi(3) = 9.425 \text{ mm}$
 c. $P_d = 25.4/m = 25.4/3 = 8.47$
 d. $P_d = 8$
 e. $a = m = 3.00 \text{ mm}$

f. $b = 1.25 \text{ mm} = 1.25(3) = 3.750 \text{ mm}$
 g. $c = 0.25 \text{ mm} = 0.25(3) = 0.750 \text{ mm}$
 h. $h_t = a + b = 2.25 \text{ mm} = 6.750 \text{ mm}$
 i. $h_k = 2a = 2 \text{ mm} = 6.00 \text{ mm}$
 j. $t = P_c/2 = \pi m/2 = 4.712 \text{ mm}$
 k. $D_o = m(N+2) = 3(36) = 108 \text{ mm}$

11. $N = 45; m = 1.25$

a. $D = mN = 1.25(45) = 56.25 \text{ mm}$
 b. $P_c = \pi m = \pi(1.25) = 3.927 \text{ mm}$
 c. $P_d = 25.4/m = 25.4/1.25 = 20.3$
 d. $P_d = 20$
 e. $a = m = 1.25 \text{ mm}$

f. $b = 1.25(m) = 1.25(1.25) = 1.563 \text{ mm}$
 g. $C = 0.25m = 0.25(1.25) = 0.313 \text{ mm}$
 h. $h_t = 2.25m = 2.25(1.25) = 2.813 \text{ mm}$
 i. $h_k = 2m = 2(1.25) = 2.500 \text{ mm}$
 j. $t = P_c/2 = \pi m/2 = 1.963 \text{ mm}$
 k. $D_o = m(N+2) = 1.25(47) = 58.75 \text{ mm}$

12. $N = 18; m = 12$

a. $D = mN = 12(18) = 216 \text{ mm}$
 b. $P_c = \pi(12) = 37.70 \text{ mm}$
 c. $P_d = 25.4/12 = 2.117$
 d. $P_d = 2$
 e. $a = m = 12 \text{ mm}$

f. $b = 1.25(12) = 15.00 \text{ mm}$
 g. $C = 0.25(12) = 3.00 \text{ mm}$
 h. $h_t = 2.25(12) = 27.00 \text{ mm}$
 i. $h_k = 2m = 24.00 \text{ mm}$
 j. $t = \pi(12)/2 = 18.85 \text{ mm}$
 k. $D_o = m(N+2) = 12(20) = 240 \text{ mm}$

13. $N = 22; m = 20$

a. $D = mN = 20(22) = 440 \text{ mm}$
 b. $P_c = \pi(20) = 62.83 \text{ mm}$
 c. $P_d = 25.4/20 = 1.27$
 d. $P_d = 1.25$
 e. $a = m = 20 \text{ mm}$

f. $b = 1.25(20) = 25.00 \text{ mm}$
 g. $C = 0.25(20) = 5.00 \text{ mm}$
 h. $h_t = 2.25(20) = 45.00 \text{ mm}$
 i. $h_k = 2m = 2(20) = 40.00 \text{ mm}$
 j. $t = \pi(20)/2 = 31.42 \text{ mm}$
 k. $D_o = 20(24) = 480 \text{ mm}$

14. $N = 20; m = 1$

a. $D = mN = 20.00 \text{ mm}$
 b. $P_c = \pi m = 3.14 \text{ mm}$
 c. $P_d = 25.4/1 = 25.4$
 d. $P_d = 24$
 e. $a = m = 1.00 \text{ mm}$

f. $b = 1.25(1) = 1.25 \text{ mm}$
 g. $C = 0.25(1) = 0.25 \text{ mm}$
 h. $h_t = 2.25(1) = 2.25 \text{ mm}$
 i. $h_k = 2(1) = 2.00 \text{ mm}$
 j. $t = \pi(1)/2 = 1.571 \text{ mm}$
 k. $D_o = (1)(22) = 22.00 \text{ mm}$

15. $N = 180; m = 0.4$

a. $D = 0.4(180) = 72.00 \text{ mm}$
 b. $P_c = \pi(0.4) = 1.26 \text{ mm}$
 c. $P_d = 25.4/0.4 = 63.5$
 d. $P_d = 64$
 e. $a = m = 0.40 \text{ mm}$

f. $b = 1.25(0.4) = 0.50 \text{ mm}$
 g. $C = 0.25(0.4) = 0.10 \text{ mm}$
 h. $h_t = 2.25(0.4) = 0.900 \text{ mm}$
 i. $h_k = 2(0.4) = 0.80 \text{ mm}$
 j. $t = \pi(0.4)/2 = 0.628 \text{ mm}$
 k. $D_o = 0.4(182) = 72.80 \text{ mm}$

16. $N = 28 ; m = 1.5$

a. $D = mN = 1.5(28) = 42.00 \text{ mm}$
 b. $P_c = \pi m = \pi(1.5) = 4.71 \text{ mm}$
 c. $P_d = 25.4/1.5 = 16.93$
 d. $P_d = 16$
 e. $a = m = 1.50 \text{ mm}$

f. $b = 1.25(1.5) = 1.875 \text{ mm}$
 g. $C = 0.25(1.5) = 0.375 \text{ mm}$
 h. $h_t = 2.25(1.5) = 3.375 \text{ mm}$
 i. $h_k = 2(1.5) = 3.00 \text{ mm}$
 j. $t = \pi(1.5)/2 = 2.36 \text{ mm}$
 k. $D_o = 1.5(30) = 45.00 \text{ mm}$

17. $N = 28 ; m = 0.8$

a. $D = 0.8(28) = 22.40 \text{ mm}$
 b. $P_c = \pi(0.8) = 2.51 \text{ mm}$
 c. $P_d = 25.4/0.8 = 31.75$
 d. $P_d = 32$
 e. $a = m = 0.80 \text{ mm}$

f. $b = 1.25(0.8) = 1.00 \text{ mm}$
 g. $C = 0.25(0.8) = 0.20 \text{ mm}$
 h. $h_t = 2.25(0.8) = 1.80 \text{ mm}$
 i. $h_k = 2(0.8) = 1.60 \text{ mm}$
 j. $t = \pi(0.8)/2 = 1.257 \text{ mm}$
 k. $D_o = 0.8(30) = 24.00 \text{ mm}$

18. BACKLASH - SEE P. 389.

19. GEAR FROM PROB. 1: $P_d = 12$: BACKLASH 0.006 TO 0.009 IN
 GEAR FROM PROB. 12: $m = 12$: BACKLASH 0.52 TO 0.82 mm
 DEPENDING ON CENTER DISTANCE.

Velocity Ratio

20. a. $C = \frac{N_p + N_g}{2P_d} = \frac{18 + 64}{2(8)} = 5.125 \text{ in.}$

b. $VR = N_g/N_p = 64/18 = 3.556$

c. $n_g = n_p (N_p/N_g) = 2450(18/64) = 689 \text{ RPM}$

d. $N_g = \frac{\pi d n_p}{12} = \frac{\pi N_p m_p}{12 P_d} = \frac{\pi(18)(2450)}{12(8)} = 1443 \text{ ft/min}$

21. a. $C = (20 + 92)/2(4) = 14.000 \text{ in.}$
 b. $VR = 92/20 = 4.60$

c. $n_g = 225(20/92) = 48.9 \text{ RPM}$

d. $N_g = \frac{\pi(20)(225)}{12(4)} = 294.5 \text{ ft/min}$

22. a. $C = (30 + 68)/2(20) = 2.450 \text{ in.}$
 b. $VR = 68/30 = 2.267$

c. $n_g = 850(30/68) = 375 \text{ RPM}$

d. $N_g = \frac{\pi(30)(850)}{12(20)} = 334 \text{ ft/min}$

23. a. $C = (40 + 250)/2(64) = 2.266 \text{ in.}$

b. $VR = 250/40 = 6.25$

c. $n_g = 3450(40/250) = 552 \text{ RPM}$

d. $N_g = \frac{\pi(40)(3450)}{12(64)} = 565 \text{ ft/min}$

24. a. $C = (24 + 88)/2(12) = 4.667 \text{ IN.}$ | c. $M_G = 1750(24/88) = 477 \text{ RPM}$
 b. $VR = 88/24 = 3.667$ | d. $N_c = \frac{\pi(24)(1750)}{12(12)} = 916 \text{ ft/min}$

25. a. $C = (N_G + N_P)m/2 = (68 + 22)(2)/2 = 90.00 \text{ mm}$
 b. $VR = N_G/N_P = 68/22 = 3.091$
 c. $M_G = n_P \cdot N_P/N_G = 1750(22/68) = 566 \text{ RPM}$
 d. $N_c = R\omega = \frac{D\omega}{2} = \left[\frac{m N_P}{2} \frac{m_P}{1} \right] \frac{(mm)(mm)}{\text{min}} \times \frac{2\pi \text{ RAD}}{180} \times \frac{1 \text{ min}}{60 \text{ s}} \times \frac{m}{10^3 \text{ mm}}$
 $N_c = \frac{m N_P n_P}{19099} \text{ m/s} = \frac{(2)(22)(1750)}{19099} = 4.03 \text{ m/s}$

26. a. $C = (48 + 18)(0.8)/2 = 26.40 \text{ mm}$ | d. $N_c = \frac{(0.8)(18)(1150)}{19099}$
 b. $VR = 48/18 = 2.667$ | $N_c = 0.867 \text{ m/s}$
 c. $M_G = 1150(18/48) = 431 \text{ RPM}$

27. a. $C = (45 + 36)(4)/2 = 162 \text{ mm}$ | d. $N_c = \frac{(4)(36)(150)}{19099} = 1.13 \text{ m/s}$
 b. $VR = 45/36 = 1.25$
 c. $M_G = 150(36/45) = 120 \text{ RPM}$

28. a. $C = (36 + 15)(12)/2 = 306 \text{ mm}$ | d. $N_c = \frac{(12)(15)(480)}{19099} = 4.52 \text{ m/s}$
 b. $VR = 36/15 = 2.40$
 c. $M_G = 480(15/36) = 200 \text{ RPM}$

Errors in statements for Problems 29 - 32

29. PINION AND GEAR CANT HAVE DIFFERENT PITCHES

30. $C = \frac{N_P + N_G}{2P_d} = \frac{18 + 82}{2(6)} = 8.333 \text{ IN.}$ GIVEN C IS INACCURATE BY 0.033 IN

31. TOO FEW TEETH IN THE PINION, ASSUMING 20° F.O. TEETH INTERFERENCE WOULD OCCUR.

32. $C = \frac{N_P + N_G}{2P_d} = \frac{24 + 45}{2(16)} = 2.156 \text{ IN.}$ D_o CANNOT BE USED TO FIND C.

Housing Dimensions

33. HOUSING MUST CLEAR ADDENDUM CIRCLE OF ALL GEARS BY 0.10 IN/SIDE
 $a = 1/P_d = 1/8 = 0.125 \text{ IN.}$; $D_{oG} = (N_G + 2)/P_d = 66/8 = 8.25 \text{ IN}$
 $Y = 8.25 \text{ IN} + 2(0.10) = 8.45 \text{ IN.}$
 $X = d + D + 2a + 2(0.10) = \frac{N_P}{P_d} + \frac{N_G}{P_d} + \frac{2}{P_d} + 2(0.1) = 2.250 + 8.000 + .25 + .20$
 $X = 10.700 \text{ IN.}$

$$34. \quad D_{og} = (N_g + 2)/P_d = 252/64 = 3.938 \text{ in}; \quad Y = 3.938 + 0.2 = 4.138 \text{ in} = Y$$

$$X = d + D + 2a + 2(0.1) = \frac{40}{64} + \frac{250}{64} + \frac{2}{64} + 0.20 = 4.763 \text{ in} = X$$

$$35. \quad D_{og} = (N_g + 2)m = 50(0.8) = 40.0 \text{ mm}; \quad Y = 40.0 + 2(2 \text{ mm}) = 44.00 \text{ mm}$$

$$X = d + D + 2a + 2(2) = m N_p + m N_g + 2m + 2(2)$$

$$X = 0.8(18) + 0.8(48) + 2(0.8) + 4.0 = 58.40 \text{ mm} = X$$

$$36. \quad D_{og} = 47(4) = 188 \text{ mm}; \quad Y = 188 + 2(2) = 192 \text{ mm}$$

$$X = d + D + 2a + 2(2) = m N_p + m N_g + 2m + 4 =$$

$$X = 144 + 180 + 8 + 4 = 336 \text{ mm} = X$$

Gear Trains - Analysis

$$37. \quad TV = - \frac{N_B}{N_A} \cdot \frac{N_D}{N_C} \cdot \frac{N_F}{N_E} = - \frac{42}{18} \cdot \frac{54}{18} \cdot \frac{54}{24} = -15.75 = M_{in}/M_{out}$$

$$M_{out} = M_{in}/TV = 1750 \text{ RPM}/(-15.75) = \underline{-111 \text{ RPM CCW}}$$

$$38. \quad TV = - \frac{N_B}{N_A} \cdot \frac{N_C}{N_D} \cdot \frac{N_E}{N_F} = - \frac{68}{22} \cdot \frac{68}{25} = -8.407 = M_{in}/M_{out}$$

$$M_{out} = M_{in}/TV = 1750 \text{ RPM}/-8.407 = \underline{-208 \text{ RPM CCW}}$$

$$39. \quad TV = + \frac{D_B}{D_A} \cdot \frac{D_D}{D_C} \cdot \frac{D_F}{D_E} \cdot \frac{N_H}{N_G} = \frac{2.875}{1.250} \cdot \frac{2.375}{1.125} \cdot \frac{2.250}{1.500} \cdot \frac{30}{18} = 12.139$$

$$\begin{array}{l} D_A = N_A/P_d = 20/16 = 1.250 \text{ in} \\ D_D = N_D/P_d = 38/16 = 2.375 \\ D_E = N_E/P_d = 18/12 = 1.500 \end{array} \quad \left| \quad M_{out} = \frac{M_{in}}{TV} = \frac{1750}{12.139} = \underline{144 \text{ RPM CW}} \right.$$

$$40. \quad TV = + \frac{N_B}{N_A} \cdot \frac{N_D}{N_C} = \frac{24}{80} \cdot \frac{18}{60} = +0.09$$

$$M_{out} = M_{in}/TV = 1750/0.09 = \underline{19444 \text{ RPM CW}}$$

Helical Gears

41 HELICAL GEAR $P_d = 8$, $\phi_t = 14\frac{1}{2}^\circ$, $N = 45$ TEETH, $F = 2.00$ IN
HELIX ANGLE $= \psi = 30^\circ$.

$$\text{CIRCULAR PITCH} = \underline{p} = \pi/P_d = \pi/8 = \underline{0.3927 \text{ IN.}}$$

$$\text{NORMAL CIRCULAR PITCH} = \underline{p_n} = p \cos \psi = (0.3927) \cos(30^\circ) = \underline{0.340 \text{ IN.}}$$

$$\text{NORMAL DIAMETRAL PITCH} = \underline{P_{nd}} = P_d / \cos \psi = 8 / \cos(30^\circ) = \underline{9.238}$$

$$\text{AXIAL PITCH} = \underline{P_x} = \frac{\pi}{P_d \tan \psi} = \frac{\pi}{8 \tan(30^\circ)} = \underline{0.680 \text{ IN.}}$$

$$\text{PITCH DIAMETER} = \underline{D_g} = N/P_d = 45/8 = \underline{5.625 \text{ IN.}}$$

$$\text{NORMAL PRESSURE ANGLE} = \underline{\phi_m} = \tan^{-1} [\tan \phi_t \cos \psi]$$

$$\underline{\phi_m} = \tan^{-1} [\tan(14.5^\circ) \cos(30^\circ)] = \underline{12.62^\circ}$$

$$F/P_x = 2.00 \text{ IN} / 0.680 \text{ IN} = \underline{2.94 \text{ AXIAL PITCHES IN FACEWIDTH}}$$

42 HELICAL GEAR $N = 48$, $P_{nd} = 12$, $\phi_n = 20^\circ$, $F = 1.50$ IN, $\psi = 45^\circ$.

$$p = \pi/P_d - \text{BUT } P_d = P_{nd} \cos \psi = 12 \cdot \cos(45^\circ) = \underline{8.485}$$

$$\underline{p} = \pi/8.485 = \underline{0.370 \text{ IN.}} ; \underline{p_n} = p \cos \psi = \pi/P_{nd} = \pi/12 = \underline{0.2618 \text{ IN.}}$$

$$\underline{P_x} = p / \tan \psi = \frac{0.370 \text{ IN.}}{\tan 45^\circ} = \underline{0.370 \text{ IN.}} ; \underline{D_g} = N/P_d = 48/8.485 = \underline{5.657 \text{ IN.}}$$

$$\underline{\phi_t} = \tan^{-1} \left[\frac{\tan \phi_m}{\cos \psi} \right] = \tan^{-1} \left[\frac{\tan 20^\circ}{\cos 45^\circ} \right] = \underline{27.2^\circ}$$

$$F/P_x = 1.500 \text{ IN} / 0.370 \text{ IN} = \underline{4.05 \text{ AXIAL PITCHES IN FACE WIDTH}}$$

43 HELICAL GEAR $N = 36$, $P_d = 6$, $\phi_t = 14\frac{1}{2}^\circ$, $\psi = 45^\circ$, $F = 1.00$ IN

$$\underline{p} = \pi/P_d = \pi/6 = \underline{0.5236 \text{ IN.}} ; \underline{p_n} = p \cos \psi = \frac{\pi}{6} \cdot \cos(45^\circ) = \underline{0.370 \text{ IN.}}$$

$$\underline{P_{nd}} = P_d / \cos \psi = 6 / \cos 45^\circ = \underline{8.485} ; \underline{P_x} = \frac{\pi}{P_d \tan \psi} = \frac{\pi}{6 \cdot \tan 45^\circ} = \underline{0.5236 \text{ IN.}}$$

$$\underline{D} = N/P_d = 36/6 = \underline{6.000 \text{ IN.}} ; \underline{\phi_m} = \tan^{-1} [\tan \phi_t \cos \psi] = \underline{10.36^\circ}$$

$$F/P_x = 1.00 \text{ IN} / 0.5236 \text{ IN} = \underline{1.91 \text{ AXIAL PITCHES IN FACE WIDTH (LOW)}}$$

44 HELICAL GEAR $N = 72$; $P_{nd} = 24$; $\phi_n = 14\frac{1}{2}^\circ$; $F = 0.25$ IN, $\psi = 45^\circ$.

$$p = \pi/P_d, \text{ BUT } P_d = P_{nd} \cos \psi = 24 \cos 45^\circ = \underline{16.97}$$

$$\underline{p} = \pi/16.97 = \underline{0.1851 \text{ IN.}} ; \underline{p_n} = p \cos \psi = 0.1851 \text{ IN.} \cdot \cos 45^\circ = \underline{0.1309 \text{ IN.}}$$

$$\underline{P_x} = p / \tan \psi = 0.1851 / \tan 45^\circ = \underline{0.1851 \text{ IN.}} ; \underline{D_g} = N/P_d = 72/16.97 = \underline{4.243 \text{ IN.}}$$

$$\underline{\phi_t} = \tan^{-1} \left[\frac{\tan \phi_n}{\cos \psi} \right] = \tan^{-1} \left[\frac{\tan(14.5^\circ)}{\cos 45^\circ} \right] = \underline{20.0^\circ} ; F/P_x = 0.25 / 0.185 = \underline{1.35}$$

LOW

SEE PROBLEM 49 ON NEXT PAGE FOR FORMULAS AND SYMBOLS

BEVEL GEAR GEOMETRY	
PROBLEM 46	EXAMPLE PROBLEM 47
GIVEN DATA	GIVEN DATA
No of teeth in pinion 15	No of teeth in pinion 25
No of teeth in gear 45	No of teeth in gear 50
Diametral pitch 6	Diametral pitch 10
Pressure angle 20 degrees	Pressure angle 20 degrees
COMPUTED VALUES	COMPUTED VALUES
Gear ratio 3.000	Gear ratio 2.000
Pitch diameter: Pinion 2.500 in	Pitch diameter: Pinion 2.500 in
Pitch diameter: Gear 7.500 in	Pitch diameter: Gear 5.000 in
Pitch cone angle: Pinion 18.435 degrees	Pitch cone angle: Pinion 26.565 degrees
Pitch cone angle: Gear 71.565 degrees	Pitch cone angle: Gear 63.435 degrees
Outer cone distance 3.953 in	Outer cone distance 2.795 in
Nominal face width 1.186 in	Nominal face width 0.839 in
Maximum face width (a) 1.318 in	Maximum face width (a) 0.932 in
Maximum face width (b) 1.667 in	Maximum face width (b) 1.000 in
INPUT Face width 1.250 in	INPUT Face width 0.900 in
Mean cone distance 3.328 in	Mean cone distance 2.345 in
Ratio A_m/A_o 0.842	Ratio A_m/A_o 0.839
Mean circular pitch 0.441 in	Mean circular pitch 0.264 in
mean working depth 0.281 in	mean working depth 0.168 in
Clearance 0.035 in	Clearance 0.021 in
Mean whole depth 0.316 in	Mean whole depth 0.189 in
mean addendum factor 0.242	mean addendum factor 0.283
Gear mean addendum 0.068 in	Gear mean addendum 0.047 in
Pinion mean addendum 0.213 in	Pinion mean addendum 0.120 in
Gear mean dedendum 0.248 in	Gear mean dedendum 0.141 in
Pinion mean dedendum 0.103 in	Pinion mean dedendum 0.068 in
Gear dedendum angle 4.257 degrees	Gear dedendum angle 3.450 degrees
Pinion dedendum angle 1.774 degrees	Pinion dedendum angle 1.670 degrees
Gear outer addendum 0.087 in	Gear outer addendum 0.061 in
Pinion outer addendum 0.259 in	Pinion outer addendum 0.148 in
Gear outside diameter 7.555 in	Gear outside diameter 5.054 in
Pinion outside diameter 2.992 in	Pinion outside diameter 2.764 in

NOTE: Maximum face width is the smallest of (a) or (b)

Given: $N_P = 18$; $N_G = 72$; $P_d = 12$; 20° pressure angle.

Computed values:

Gear ratio	$m_G = N_G/N_P = 72/18 = 4.000$
Pitch diameter: Pinion	$d = N_P/P_d = 18/12 = 1.500$ in
Pitch diameter: Gear	$D = N_G/P_d = 72/12 = 6.000$ in
Pitch cone angle: Pinion	$\gamma = \tan^{-1}(N_P/N_G) = \tan^{-1}(18/72) = 14.03^\circ$
Pitch cone angle: Gear	$\Gamma = \tan^{-1}(N_G/N_P) = \tan^{-1}(72/18) = 75.96^\circ$
Outer cone distance	$A_o = 0.5D/\sin(\Gamma) = 0.5(6.00 \text{ in})/\sin(75.96^\circ) = 3.092$ in

Face width must be specified: $F = 0.800$ in Based on the following guidelines:

Nominal face width: $F_{nom} = 0.30 A_o = 0.30(3.092 \text{ in}) = 0.928$ in

Maximum face width: $F_{max} = A_o/3 = (3.092 \text{ in})/3 = 1.031$ in

or $F_{max} = 10/P_d = 10/12 = 0.833$ in

Mean cone distance $A_m = A_{mG} = A_o - 0.5F = 3.092 \text{ in} - 0.5(0.80 \text{ in}) = 2.692$ in

Ratio $(A_m/A_o) = (2.692/3.092) = 0.871$ [This ratio occurs in several following calculations]

Mean circular pitch $p_m = (\pi/P_d)(A_m/A_o) = (\pi/12)(0.871) = 0.228$ in

Mean working depth $h = (2.00/P_d)(A_m/A_o) = (2.00/12)(0.871) = 0.145$ in

Clearance $c = 0.125h = 0.125(0.145 \text{ in}) = 0.018$ in

Mean whole depth $h_m = h + c = 0.145 \text{ in} + 0.018 \text{ in} = 0.163$ in

Mean addendum factor $c_1 = 0.210 + 0.290/(m_G)^2 = 0.210 + 0.290/(4.00)^2 = 0.228$

Gear mean addendum $a_G = c_1 h = (0.228)(0.145 \text{ in}) = 0.033$ in

Pinion mean addendum $a_P = h - a_G = 0.145 \text{ in} - 0.033 \text{ in} = 0.112$ in

Gear mean dedendum $b_G = h_m - a_G = 0.163 \text{ in} - 0.033 \text{ in} = 0.130$ in

Pinion mean dedendum $b_P = h_m - a_P = 0.163 \text{ in} - 0.112 \text{ in} = 0.051$ in

Gear dedendum angle $\delta_G = \tan^{-1}(b_G/A_{mG}) = \tan^{-1}(0.130/2.692) = 2.76^\circ$

Pinion dedendum angle $\delta_P = \tan^{-1}(b_P/A_{mG}) = \tan^{-1}(0.051/2.692) = 1.09^\circ$

Gear outer addendum $a_{oG} = a_G + 0.5F \tan \delta_P$

$a_{oG} = (0.033 \text{ in}) + (0.5)(0.80 \text{ in})\tan(1.09^\circ) = 0.0406$ in

Pinion outer addendum $a_{oP} = a_P + 0.5F \tan \delta_G$

$a_{oP} = (0.112 \text{ in}) + (0.5)(0.80 \text{ in})\tan(2.76^\circ) = 0.1313$ in

Gear outside diameter $D_o = D + 2a_{oG} \cos \Gamma$

$D_o = 6.000 \text{ in} + 2(0.0406 \text{ in})\cos(75.96^\circ) = 6.020$ in

Pinion outside diameter $d_o = d + 2a_{oP} \cos \gamma$

$d_o = 1.500 \text{ in} + 2(0.1313 \text{ in})\cos(14.04^\circ) = 1.755$ in

BEVEL GEAR GEOMETRY

PROBLEM- 49

GIVEN DATA

No. of teeth in pinion	18
No. of teeth in gear	72
Diametral pitch	12
Pressure angle	20 degrees

COMPUTED VALUES

Gear ratio	4.000
Pitch diameter: Pinion	1.500 in
Pitch diameter: Gear	6.000 in
Pitch cone angle: Pinion	14.036 degrees
Pitch cone angle: Gear	75.964 degrees
Outer cone distance	3.092 in

Nominal face width	0.928 in
Maximum face width (a)	1.031 in
Maximum face width (b)	0.833 in

INPUT Face width 0.800 in

Mean cone distance	2.692 in
Ratio A_m/A_o	0.871
Mean circular pitch	0.228 in
mean working depth	0.145 in
Clearance	0.018 in
Mean whole depth	0.163 in
mean addendum factor	0.228
Gear mean addendum	0.033 in
Pinion mean addendum	0.112 in
Gear mean dedendum	0.130 in
Pinion mean dedendum	0.051 in
Gear dedendum angle	2.767 degrees
Pinion dedendum angle	1.090 degrees
Gear outer addendum	0.041 in
Pinion outer addendum	0.131 in
Gear outside diameter	6.020 in
Pinion outside diameter	1.755 in

PROBLEM- 50

GIVEN DATA

No. of teeth in pinion	16
No. of teeth in gear	64
Diametral pitch	32
Pressure angle	20 degrees

COMPUTED VALUES

Gear ratio	4.000
Pitch diameter: Pinion	0.500 in
Pitch diameter: Gear	2.000 in
Pitch cone angle: Pinion	14.036 degrees
Pitch cone angle: Gear	75.964 degrees
Outer cone distance	1.031 in

Nominal face width	0.309 in
Maximum face width (a)	0.344 in
Maximum face width (b)	0.313 in

INPUT Face width 0.300 in

Mean cone distance	0.881 in
Ratio A_m/A_o	0.854
Mean circular pitch	0.084 in
mean working depth	0.053 in
Clearance	0.007 in
Mean whole depth	0.060 in
mean addendum factor	0.228
Gear mean addendum	0.012 in
Pinion mean addendum	0.041 in
Gear mean dedendum	0.048 in
Pinion mean dedendum	0.019 in
Gear dedendum angle	3.113 degrees
Pinion dedendum angle	1.227 degrees
Gear outer addendum	0.015 in
Pinion outer addendum	0.049 in
Gear outside diameter	2.007 in
Pinion outside diameter	0.596 in

NOTE: Maximum face width is the smallest of (a) or (b)

BEVEL GEAR GEOMETRY

PROBLEM 51

GIVEN DATA

No. of teeth in pinion	12
No. of teeth in gear	36
Diametral pitch	48
Pressure angle	20 degrees

COMPUTED VALUES

Gear ratio	3.000
Pitch diameter: Pinion	0.250 in
Pitch diameter: Gear	0.750 in
Pitch cone angle: Pinion	18.435 degrees
Pitch cone angle: Gear	71.565 degrees
Outer cone distance	0.395 in

Nominal face width	0.119 in
Maximum face width (a)	0.132 in
Maximum face width (b)	0.208 in

INPUT Face width 0.125 in

Mean cone distance	0.333 in
Ratio A_m/A_o	0.842
Mean circular pitch	0.055 in
mean working depth	0.035 in
Clearance	0.004 in
Mean whole depth	0.039 in
mean addendum factor	0.242
Gear mean addendum	0.008 in
Pinion mean addendum	0.027 in
Gear mean dedendum	0.031 in
Pinion mean dedendum	0.013 in
Gear dedendum angle	5.316 degrees
Pinion dedendum angle	2.217 degrees
Gear outer addendum	0.011 in
Pinion outer addendum	0.032 in
Gear outside diameter	0.757 in
Pinion outside diameter	0.311 in

EXAMPLE PROBLEM 8-8

GIVEN DATA

No. of teeth in pinion	16
No. of teeth in gear	48
Diametral pitch	8
Pressure angle	20 degrees

COMPUTED VALUES

Gear ratio	3.000
Pitch diameter: Pinion	2.000 in
Pitch diameter: Gear	6.000 in
Pitch cone angle: Pinion	18.435 degrees
Pitch cone angle: Gear	71.565 degrees
Outer cone distance	3.162 in

Nominal face width	0.949 in
Maximum face width (a)	1.054 in
Maximum face width (b)	1.250 in

INPUT Face width 1.000 in

Mean cone distance	2.662 in
Ratio A_m/A_o	0.842
Mean circular pitch	0.331 in
mean working depth	0.210 in
Clearance	0.026 in
Mean whole depth	0.237 in
mean addendum factor	0.242
Gear mean addendum	0.051 in
Pinion mean addendum	0.159 in
Gear mean dedendum	0.186 in
Pinion mean dedendum	0.077 in
Gear dedendum angle	3.992 degrees
Pinion dedendum angle	1.663 degrees
Gear outer addendum	0.065 in
Pinion outer addendum	0.194 in
Gear outside diameter	6.041 in
Pinion outside diameter	2.369 in

NOTE: Maximum face width is the smallest of (a) or (b)

Wormgearing

52

WORM GEARING : $D_W = 1.250 \text{ IN.}$, $N_W = 1$, $P_d = 10$; $\phi_n = 14.5^\circ$

$N_G = 40$; $F = 0.625 \text{ IN.}$

SINGLE THREAD

LEAD = AXIAL PITCH = CIRCULAR PITCH = $\frac{\pi}{P_d} = \frac{\pi}{10} = 0.3142 \text{ IN.}$

LEAD ANGLE = $\lambda = \tan^{-1} \left(\frac{L}{\pi D_W} \right) = \tan^{-1} \left(\frac{0.3142}{\pi(1.250)} \right) = 4.57^\circ$

ADDENDUM = $a = \frac{1}{P_d} = \frac{1}{10} = 0.100 \text{ IN.}$; DEDENDUM = $\frac{1.157}{P_d} = 0.1157 \text{ IN.}$

WORM OUTSIDE DIA. = $D_{oW} = D_W + 2a = 1.250 + 2(0.100) = 1.450 \text{ IN.}$

WORM ROOT DIA. = $D_{rW} = D_W - 2b = 1.250 - 2(0.1157) = 1.0186 \text{ IN.}$

GEAR PITCH DIA. = $D_G = N_G / P_d = 40 / 10 = 4.000 \text{ IN.}$

CENTER DISTANCE = $C = (D_G + D_W) / 2 = (4.000 + 1.250) / 2 = 2.625 \text{ IN.}$

VELOCITY RATIO = $VR = N_G / N_W = 40 / 1 = 40$

NOTE: On the following two pages are the results of Problems 52-57 giving pertinent geometric properties of worms and wormgears and their velocity ratios. The detailed calculations follow the pattern illustrated above for Problem 52. The equations come from Section 8-9, Equations 8-20 to 8-25.

Compare the results to discern how variations in geometry such as diametral pitch and the number of threads in the worm affect the overall results. This is especially pertinent to Problem 53 in which three different designs for worm/wormgear sets provide the same velocity ratio. The single threaded worm produces the smallest center distance and overall size of the reducer. But note, also, that it has the smallest lead angle. The lead angle increases as the number of threads is increased. On the positive side, the small lead angle makes the reducer self-locking. On the negative side, the small lead angle results in lower mechanical efficiency as will be shown in Chapter 10, Section 10-11. The designer must balance these advantages and disadvantages for each application.

WORMGEARING **PROBLEM: 52**
INPUT DATA

Worm pitch diameter = 1.250 in
Diametral pitch = 10
No. of worm threads = 1
No. of gear teeth = 40
Face width of gear = 0.625 in

COMPUTED RESULTS

Circular pitch of gear = 0.3142 in
Axial pitch of worm = 0.3142 in
Lead of the worm = 0.3142 in
Lead angle = 4.574 deg
Addendum = 0.100 in
Dedendum = 0.116 in
Worm outside diameter = 1.450 in
Worm root diameter = 1.019 in
Gear pitch diameter = 4.000 in
Center distance = 2.625 in
Velocity ratio = 40.00

WORMGEARING **PROBLEM: 53A**
INPUT DATA

Worm pitch diameter = 1.000 in
Diametral pitch = 12
No. of worm threads = 1
No. of gear teeth = 20
Face width of gear = 0.500 in

COMPUTED RESULTS

Circular pitch of gear = 0.2618 in
Axial pitch of worm = 0.2618 in
Lead of the worm = 0.2618 in
Lead angle = 4.764 deg
Addendum = 0.083 in
Dedendum = 0.096 in
Worm outside diameter = 1.167 in
Worm root diameter = 0.807 in
Gear pitch diameter = 1.667 in
Center distance = 1.333 in
Velocity ratio = 20.00

WORMGEARING **PROBLEM: 53B**
INPUT DATA

Worm pitch diameter = 1.000 in
Diametral pitch = 12
No. of worm threads = 2
No. of gear teeth = 40
Face width of gear = 0.500 in

COMPUTED RESULTS

Circular pitch of gear = 0.2618 in
Axial pitch of worm = 0.2618 in
Lead of the worm = 0.5236 in
Lead angle = 9.462 deg
Addendum = 0.083 in
Dedendum = 0.096 in
Worm outside diameter = 1.167 in
Worm root diameter = 0.807 in
Gear pitch diameter = 3.333 in
Center distance = 2.167 in
Velocity ratio = 20.00

WORMGEARING **PROBLEM: 53C**
INPUT DATA

Worm pitch diameter = 1.000 in
Diametral pitch = 12
No. of worm threads = 4
No. of gear teeth = 80
Face width of gear = 0.500 in

COMPUTED RESULTS

Circular pitch of gear = 0.2618 in
Axial pitch of worm = 0.2618 in
Lead of the worm = 1.0472 in
Lead angle = 18.435 deg
Addendum = 0.083 in
Dedendum = 0.096 in
Worm outside diameter = 1.167 in
Worm root diameter = 0.807 in
Gear pitch diameter = 6.667 in
Center distance = 3.833 in
Velocity ratio = 20.00

WORMGEARING **PROBLEM: 54**
INPUT DATA

Worm pitch diameter = 0.625 in
 Diametral pitch = 16
 No. of worm threads = 2
 No. of gear teeth = 100
 Face width of gear = 0.313 in

COMPUTED RESULTS

Circular pitch of gear = 0.1963 in
 Axial pitch of worm = 0.1963 in
 Lead of the worm = 0.3927 in
 Lead angle = 11.310 deg
 Addendum = 0.063 in
 Dedendum = 0.072 in
 Worm outside diameter = 0.750 in
 Worm root diameter = 0.480 in
 Gear pitch diameter = 6.250 in
 Center distance = 3.438 in
 Velocity ratio = 50.00

WORMGEARING **PROBLEM: 55**
INPUT DATA

Worm pitch diameter = 2.000 in
 Diametral pitch = 6
 No. of worm threads = 4
 No. of gear teeth = 72
 Face width of gear = 1.000 in

COMPUTED RESULTS

Circular pitch of gear = 0.5236 in
 Axial pitch of worm = 0.5236 in
 Lead of the worm = 2.0944 in
 Lead angle = 18.435 deg
 Addendum = 0.167 in
 Dedendum = 0.193 in
 Worm outside diameter = 2.333 in
 Worm root diameter = 1.614 in
 Gear pitch diameter = 12.000 in
 Center distance = 7.000 in
 Velocity ratio = 18.00

WORMGEARING **PROBLEM: 56**
INPUT DATA

Worm pitch diameter = 4.000 in
 Diametral pitch = 3
 No. of worm threads = 1
 No. of gear teeth = 54
 Face width of gear = 2.000 in

COMPUTED RESULTS

Circular pitch of gear = 1.0472 in
 Axial pitch of worm = 1.0472 in
 Lead of the worm = 1.0472 in
 Lead angle = 4.764 deg
 Addendum = 0.333 in
 Dedendum = 0.386 in
 Worm outside diameter = 4.667 in
 Worm root diameter = 3.229 in
 Gear pitch diameter = 18.000 in
 Center distance = 11.000 in
 Velocity ratio = 54.00

WORMGEARING **PROBLEM: 57**
INPUT DATA

Worm pitch diameter = 0.333 in
 Diametral pitch = 48
 No. of worm threads = 4
 No. of gear teeth = 80
 Face width of gear = 0.156 in

COMPUTED RESULTS

Circular pitch of gear = 0.0654 in
 Axial pitch of worm = 0.0654 in
 Lead of the worm = 0.2618 in
 Lead angle = 14.050 deg
 Addendum = 0.021 in
 Dedendum = 0.024 in
 Worm outside diameter = 0.375 in
 Worm root diameter = 0.285 in
 Gear pitch diameter = 1.667 in
 Center distance = 1.000 in
 Velocity ratio = 20.00

Gear Trains - Analysis

FOR PROBLEM 58 - ASSUME THAT THE INPUT SHAFT ROTATES CLOCKWISE.

58

$$\text{TRAIN VALUE} = TV = m_1/m_6 ; m_1 = 3450 \text{ RPM}$$

$$TV = \frac{-N_B}{N_A} \cdot \frac{N_D}{N_C} \cdot \frac{N_F}{N_E} \cdot \frac{N_H}{N_G} \cdot \frac{N_I}{N_H} = \frac{-82}{18} \cdot \frac{64}{17} \cdot \frac{110}{20} \cdot \frac{18}{18} \cdot \frac{38}{18} = -119.1$$

$$m_6 = \frac{m_1}{TV} = \frac{3450 \text{ RPM}}{-119.1} = -17.32 \text{ RPM COUNTERCLOCKWISE}$$

GEAR H IS AN IDLER. IT DOES NOT AFFECT THE TV BUT CHANGES THE DIRECTION OF THE OUTPUT SHAFT.

59

$$m_1 = 12200 \text{ RPM} ; \text{FIND } m_5 : TV = m_1/m_5$$

$$TV = \frac{N_B}{N_A} \cdot \frac{N_D}{N_C} \cdot \frac{N_F}{N_E} \cdot \frac{N_H}{N_G} = \frac{50}{12} \cdot \frac{40}{12} \cdot \frac{60}{1} \cdot \frac{72}{2} = 30000$$

$$m_5 = \frac{m_1}{TV} = \frac{12200 \text{ RPM}}{30000} = 0.4067 \text{ RPM}$$

60

$$m_1 = 6840 \text{ RPM} ; \text{FIND } m_4 : TV = m_1/m_4$$

$$TV = \frac{N_B}{N_A} \cdot \frac{N_D}{N_C} \cdot \frac{N_F}{N_E} = \frac{48}{16} \cdot \frac{48}{18} \cdot \frac{60}{12} = 40 \text{ EXACTLY}$$

$$m_4 = \frac{m_1}{TV} = \frac{6840}{40} = 171 \text{ RPM EXACTLY}$$

61

$$m_1 = 2875 \text{ RPM} ; \text{FIND } m_4 : TV = m_1/m_4$$

$$TV = \frac{N_B}{N_A} \cdot \frac{N_D}{N_C} \cdot \frac{N_F}{N_E} = \frac{100}{3} \cdot \frac{80}{2} \cdot \frac{85}{20} = 5666.7$$

$$m_4 = \frac{m_1}{TV} = \frac{2875 \text{ RPM}}{5666.7} = 0.5074 \text{ RPM}$$

Gear Trains - Kinematic Design

VELOCITY RATIO FOR GEARS <i>PROBLEM 62</i>					
DESIRED VR = 3.1416 = π					
NP	NG	NG Act	VR-Act	DIFF = Des VR - VR Act	
16	50.27	50	3.1250	0.01659	
17	53.41	53	3.1176	0.02395	
18	56.55	57	3.1667	0.02507	
19	59.69	60	3.1579	0.01630	
20	62.83	63	3.1500	0.00841	
XX	21	65.97	66	3.1429	0.00126 XX
	22	69.12	69	3.1364	0.00523
	23	72.26	72	3.1304	0.01116
	24	75.40	75	3.1250	0.01659
				Min diff =	0.00126

VELOCITY RATIO FOR GEARS <i>PROBLEM 63</i>					
DESIRED VR = 1.7321 = $\sqrt{3}$					
NP	NG	NG Actual	VR Actual	DIFF = Des VR - VR Act	
16	27.71	28	1.7500	0.01795	
17	29.44	29	1.7059	0.02617	
18	31.18	31	1.7222	0.00983	
19	32.91	33	1.7368	0.00479	
20	34.64	35	1.7500	0.01795	
	21	36.37	36	1.7143	0.01777
XX	22	38.11	38	1.7273	0.00478 XX
	23	39.84	40	1.7391	0.00708
	24	41.57	42	1.7500	0.01795
				Min diff =	0.00478

VELOCITY RATIO FOR GEARS <i>PROBLEM 64</i>					
DESIRED VR = 6.1644 = $\sqrt{38}$					
NP	NG	NG Act	VR-Act	DIFF = Des VR - VR Act	
16	98.63	99	6.1875	0.02309	
17	104.80	105	6.1765	0.01206	
XX	18	110.96	111	6.1667	0.00225 XX
	19	117.12	117	6.1579	0.00652
	20	123.29	123	6.1500	0.01441
	21	129.45	129	6.1429	0.02156
	22	135.62	136	6.1818	0.01740
	23	141.78	142	6.1739	0.00950
XX	24	147.95	148	6.1667	0.00225 XX
<i>TWO EQUAL SOLUTIONS</i>				Min diff =	0.00225

VELOCITY RATIO FOR GEARS <i>PROBLEM 65</i>					
DESIRED VR = 7.42					
NP	NG	NG Actual	VR Actual	DIFF = Des VR - VR Act	
16	118.72	119	7.4375	0.01750	
17	126.14	126	7.4118	0.00824	
18	133.56	134	7.4444	0.02444	
XX	19	140.98	141	7.4211	0.00105 XX
	20	148.40	148	7.4000	0.02000
	21	155.82	156	7.4286	0.00857
	22	163.24	163	7.4091	0.01091
	23	170.66	171	7.4348	0.01478
	24	178.08	178	7.4167	0.00333
				Min diff =	0.00105

66

DESIGN: $M_{IN} = 1800 \text{ RPM}$ $M_{OUT} = 2 \text{ RPM}$ EXACT RATIO RED'D.

$TV = 1800/2 = 900$ EXACT; USE FACTORING: $N_{MAX} = 150$

$$\begin{array}{r} 2 \overline{) 900} \\ 2 \overline{) 450} \\ 5 \overline{) 225} \\ 5 \overline{) 45} \\ 3 \overline{) 9} \\ 3 \end{array}$$

FACTORS ARE: 2 · 2 · 5 · 5 · 3 · 3

SEE TABLE 8-6 FOR INTERFERENCE DATA

FOR 20° F.D. TEETH USE $N_{MIN} = 16$ OR 17

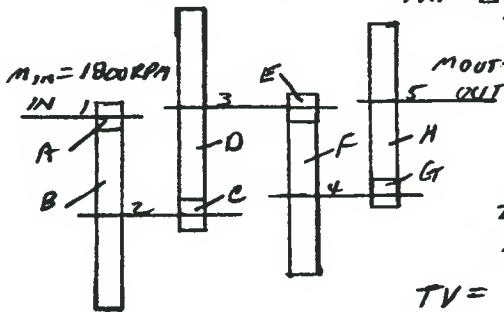
NOMINAL VR_{MAX} PER PAIR = $150/17 = 8.82$ TOO SMALL

TWO PAIRS: $(8.82)^2 = 77.8$ SMALL

THREE PAIRS: $(8.82)^3 = 687$ SMALL

FOUR PAIRS: $(8.82)^4 = 6061$ RED'D.

RECOMBINE FACTORS: $VR_1 = 6, VR_2 = 6, 1 : 3 = 5, VR_4 = 5$



$$\begin{array}{l|l} N_A = 16 & N_E = 16 \\ N_B = 96 & N_F = 80 \\ N_C = 16 & N_G = 16 \\ N_D = 96 & N_H = 80 \end{array}$$

TABLE 8-6 SAYS NO INTERFERENCE FOR $N_P = 16$ IF $N_G < 101$.

$$TV = \frac{96}{16} \frac{96}{16} \frac{80}{16} \frac{80}{16} = 900 \text{ EXACT}$$

67

DESIGN: $M_{IN} = 1800 \text{ RPM EXACT}$; $21 < M_{OUT} < 22$; $N_{MAX} = 150$ & 20° F.D.

$$TV_{MIN} = 1800/21.5 = 83.7 \quad TV_{MIN} = 1800/22 = 81.8 \quad TV_{MAX} = 1800/21 = 85.7$$

FROM TABLE 8-6, NO INTERFERENCE WITH $N_P \geq 17$ FOR 20° F.D. TEETH

VR_{MAX} PER PAIR = $150/17 = 8.82$ SMALL; TWO PAIRS $VR_{MAX} = (8.82)^2 = 77.9$ LOW

LAYOUT AS IN FIG. 8-31 IN TEXT—TRIPLE REDUCTION.

TRY EQUAL REDUCTION RATIO: $VR_1 = VR_2 \approx VR_3 = \sqrt[3]{83.7} = 4.37$

LET $N_A = N_C = N_E = 17$; LET $VR_1 = 5, VR_2 = 4$; THEN $VR_3 = 83.7/20 = 4.19$

$$N_F = (17)(4.19) = 71.2 \Rightarrow \text{SPECIFY } N_F = 71$$

$$\text{FINAL } TV = \frac{85}{17} \cdot \frac{68}{17} \cdot \frac{71}{17} = 83.53$$

$$M_{OUT} = \frac{M_{IN}}{TV} = \frac{1800}{83.53} = 21.55 \text{ RPM}$$

68

DESIGN: $M_{IN} = 3360 \text{ RPM EXACT}$; $M_{OUT} = 12 \text{ RPM EXACT}$; $N_{MAX} = 150$

20° F.D. TEETH. FROM TABLE 8-6 LET $N_{MIN} = 17$ FOR NO INTERFERENCE

VR_{MAX} PER PAIR = $150/17 = 8.82$; 2 PAIRS $VR_{MAX} = (8.82)^2 = 77.8$; 3 PAIRS = 686

$TV = 3360/12 = 280$ EXACT; USE 3 PAIRS SIMILAR TO FIG. 8-31 IN TEXT

$$\begin{array}{r} 2 \overline{) 280} \\ 2 \overline{) 140} \\ 5 \overline{) 70} \\ 2 \overline{) 14} \\ 7 \end{array}$$

2 · 2 · 5 · 2 · 7 = 280. RECOMBINE 8 · 7 · 5 = 280

$$VR_1 = 8; N_A = 17, N_B = 136$$

$$VR_2 = 7; N_C = 17, N_D = 119$$

$$VR_3 = 5; N_E = 17, N_F = 85$$

$$TV = \frac{136}{17} \cdot \frac{119}{17} \cdot \frac{85}{17} = 280 \text{ EXACTLY}$$

(OTHER DESIGNS POSSIBLE)

69

DESIGN: $M_{IN} = 4200 \text{ RPM}$ EXACTLY: $13.0 < M_{OUT} < 13.5 \text{ RPM}$: POSITIVE TV

$$TV_{MIN} = \frac{4200}{13.5} = 311.1; TV_{NOM} = \frac{4200}{13.25} = 316.98; TV_{MAX} = \frac{4200}{13.0} = 323.08$$

FROM PROB 68, 3 PAIRS REQ'D. LAYOUT IN FIG 8-31 PRODUCES A NEGATIVE TV. USE IDLER IN ANY PAIR.

TRY RESIDUAL RATIO METHOD. NOMINAL VR = $\sqrt[3]{317} = 6.82$ PER PAIR

TRY $VR_1 = 7; VR_2 = 6$; THEN $VR_3 \approx 317/42 \approx 7.55$: USE $VR_3 = 7.50$

$$FINAL TV = VR_1 \cdot VR_2 \cdot VR_3 = (7)(6)(7.50) = 315 \text{ OK}$$

IT IS PREFERRED TO PLACE HIGHER RATIOS EARLY IN THE TRAIN. LET $VR_1 = 7.5, VR_2 = 7, VR_3 = 6$.

LET $N_A = 18; N_B = 7.5(18) = 135$; LET $N_C = 17, N_D = 7(17) = 119$

LET $N_E = 17, N_F = 17(6) = 102$.

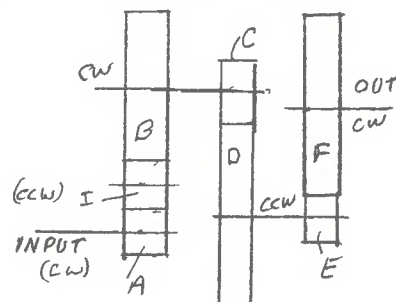
IDLER NEEDED FOR POSITIVE TRAIN

LET $N_I = 17$. PLACE IN FIRST PAIR, FINAL TRAIN VALUE:

$$TV = \frac{135}{18} \times \frac{119}{17} \times \frac{102}{17} = 315$$

$$M_{OUT} = \frac{M_{IN}}{TV} = \frac{4200}{315} = 13.33 \text{ RPM CW}$$

OK



NOTE: CHAPTER 9 GIVES INFORMATION ON SELECTION OF PD - DIAMETRAL PITCH. BECAUSE OF SPEED/TORQUE CHANGES,

$P_{d1} > P_{d2} > P_{d3}$, LARGER PD GIVES SMALLER GEARS.

THIS IS THE REASON THAT LARGER RATIOS SHOULD BE PLACED EARLIER IN THE TRAIN.

70

DESIGN: $M_{IN} = 5500 \text{ RPM}$ EXACTLY:

$$221 < M_{OUT} < 225$$

DESIGN: TWO DOUBLE REDUCTION WITH ALL EXTERNAL GEARS

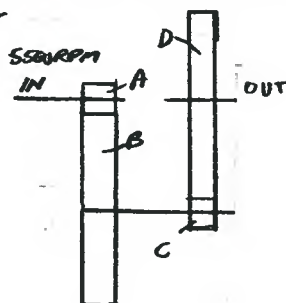
$$TV_{MAX} = \frac{5500}{221} = 24.88; TV_{MIN} = \frac{5500}{225} = 24.44; TV_{NOM} = \frac{5500}{223} = 24.66$$

NOMINAL RATIO FOR EACH PAIR = $\sqrt[4]{24.66} = 4.97$. TRY $VR_1 = 5$

THEN $VR_2 \approx 24.66/5 = 4.93$: FOR $N_F = 16, N_G = 78.9$ - USE $N_G = 79$

$$FINAL TV_1 = \frac{N_B}{N_A} \cdot \frac{N_D}{N_C} = \frac{80}{16} \cdot \frac{79}{16} = 24.6875$$

$$M_{OUT} = \frac{5500}{TV_1} = 222.78 \text{ RPM OK}$$

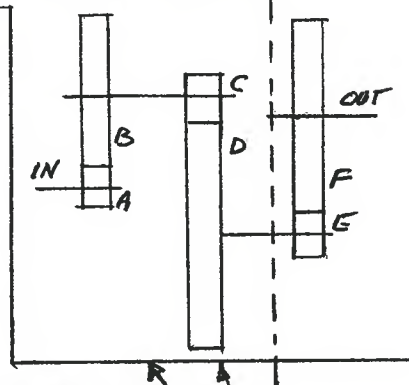


71 DESIGN: $M_{IN} = 5500 \text{ RPM}$ $13.0 < m_{OUT} < 14.0 \text{ RPM}$
 $TV_{NOM} = 5500/13.5 = 407.4$ SKETCH AS IN 70.
 MAX RATIO FOR ONE PAIR $= 150/17 = 8.82$
 TWO PAIRS - MAX $= 77.85$; THREE PAIRS 687 - OK
 NOMINAL RATIO PER PAIR: $\sqrt[3]{407.4} = 7.41$
 TRY $VR_1 = 8, VR_2 = 8$ - BUT USE HUNTING TOOTH APPROACH.
 $VR = 8$: $N_A = 17, N_B = 17(8) = 136$; USE $N_B = 135$
 SAME FOR N_C, N_D .
 $(VR_1)(VR_2) = \left(\frac{135}{17}\right)^2 = 7.94^2 = 63.06$
 RESIDUAL RATIO: $407.4/63.06 = 6.46 = N_F/N_E$
 LET $N_E = 17$; $N_F = 6.46(17) = 109.82 \Rightarrow$ USE 110 TEETH
 FINAL $TV = \frac{135}{17} \times \frac{135}{17} \times \frac{110}{17} = 408.05$
 FINAL OUTPUT SPEED $= 5500/408.05 = 13.48 \text{ RPM} - \text{OK}$

72 DESIGN: $M_{IN} = 1750$; $146 < m_{OUT} < 150$
 $TV_{NOM} = 1750/148 = 11.82$
 LET $VR_1 = N_B/N_A = 75/18 = 4.167$
 RESIDUAL RATIO $= 11.82/4.167 = 2.837$
 LET $N_C = 18$: $N_D = 18(2.837) = 51.06 \Rightarrow 51$
 $N_A = 18, N_B = 75, N_C = 18, N_D = 51$

$m_{OUT} = 1750 \times \frac{18}{75} \times \frac{18}{51} = 148.2 \text{ RPM} \text{ OK}$

SKETCH SAME AS 71 WITH ONLY TWO PAIRS
 [THESE RESULTS USED IN PROBLEM 9-74.]



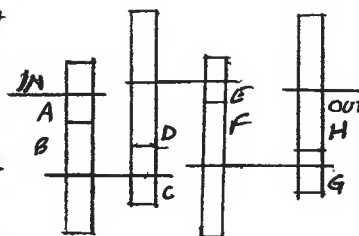
73 DESIGN: $M_{IN} = 850 \text{ RPM}$; $40 < m_{OUT} < 44$; USE 2 PAIRS
 $TV_{NOM} = 850/42 = 20.24$, LET $VR_1 = N_B/N_A = 81/18 = 4.50$
 RESIDUAL RATIO $= VR_2 = 20.24/4.50 = 4.50$; $N_C = 18, N_D = 81$
 $m_{OUT} = 850 \times \frac{18}{81} \times \frac{18}{81} = 41.98 \text{ RPM} \text{ OK}$
 [THESE RESULTS USED IN PROBLEM 9-25.]

74 DESIGN: USE TWO PAIRS: $M_{IN} = 3000 \text{ RPM}$; $548 < m_{OUT} < 552$
 $TV_{NOM} = 3000/550 = 5.4545$; LET $VR_1 = VR_2 = \sqrt{5.4545} = 2.335$
 LET $N_A = 15$; $N_B = 15(2.335) = 35.03 \Rightarrow 35$. LET $N_C = 15, N_D = 35$
 $m_{OUT} = 3000 \times \frac{15}{35} \times \frac{15}{35} = 551 \text{ RPM} \text{ OK}$
 [THESE RESULTS USED IN PROBLEM 9-26.]

75

DESIGN: $M_{IN} = 3600 \text{ RPM}$ $3.0 < M_{OUT} < 5.0$ $TV_{NOM} = 3600/4.0 = 900$; USE 4 PAIRS

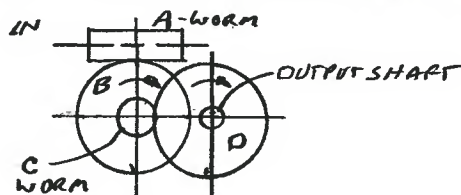
FACTORING: $2 \overline{) 900}$
 $2 \overline{) 450}$
 $5 \overline{) 225}$
 $5 \overline{) 45}$
 $3 \overline{) 9}$
 3

USE $VR_1 = 6 = 96/16 = N_B/N_A$ $VR_2 = 6 = 96/16 = N_D/N_C$ $VR_3 = 5 = 80/16 = N_F/N_E$ $VR_4 = 5 = 80/16 = N_H/N_G$ 

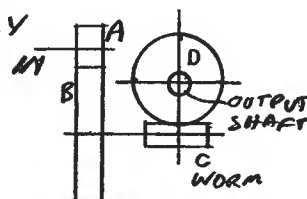
ALTERNATE SOLUTION USING HUNTING TOOTH:

LET $N_A = N_C = N_E = N_G = 16$. LET $N_B = N_D = 95$. LET $N_F = 81$ $VR_1 = VR_2 = 95/16 = 5.9375$; $VR_3 = 81/16 = 5.0625$ $VR_1 \times VR_2 \times VR_3 = 178.47$. RESIDUAL RATIO $= 900/178.47 = 5.043$ LET $N_H = 81$. $VR_4 = VR_3 = 81/16 = 5.0625$ TOTAL $TV = (178.47)(5.0625) = 903.5$ FINAL $M_{OUT} = 3600/903.5 = 3.984 \text{ RPM}$ OK

76

DESIGN: $M_{IN} = 3600$ $3.0 < M_{OUT} < 5.0 \text{ RPM}$ $TV_{NOM} = 3600/4.0 = 900$; USE TWO PAIRS OF WORM/WORM GEARS $N_A = N_C = 1$; $N_B = N_D = 30$ 

77

DESIGN: $M_{IN} = 1800 \text{ RPM}$ $M_{OUT} = 8.0 \text{ EXACTLY}$ LET $VR_2 = 50$ - WORM GEAR DRIVE. $TV = 1800/8 = 225$; $VR_1 = 225/50 = 4.50$
HELICAL GEARSLET $N_A = 16$, $N_B = 72$, $N_C = 1$, $N_D = 50$ $M_{OUT} = 1800 \times \frac{16}{72} \times \frac{1}{50} = 8.0 \text{ RPM}$ 

78

DESIGN: $M_{IN} = 3360 \text{ RPM}$. $M_{OUT} = 12.0 \text{ EXACTLY}$,
USE TWO PAIRS OF WORM GEAR DRIVES AS IN PROBLEM 76.
LET $VR_1 = 20$, $VR_2 = 14$: $N_A = 2$, $N_B = 40$, $N_C = 2$, $N_D = 28$.

79. DESIGN! $m_{IN} = 4200 \text{ RPM}$, $13.0 < m_{OUT} < 13.5$
 USE COMBINED HELICAL WITH WORM GEAR AS IN PROBLEM 77.
 LET $VR_2 = 50$. WORM GEAR DRIVE, $N_C = 1, N_D = 50$

$$TV_{NOM} = 4200 / 13.25 = 316.98. VR_1 = 316.98 / 50 = 6.34$$

$$\text{LET } N_A = 18. N_B = 18(6.34) = 114.1 \Rightarrow \text{USE } 114 = N_B$$

$$\text{FINAL OUTPUT SPEED} = 4200 \times \frac{18}{114} \times \frac{1}{50} = \underline{13.26 \text{ RPM OK}}$$

80. DESIGN! $m_{IN} = 5500 \text{ RPM}$ $13.0 < m_{OUT} < 14.0 \text{ RPM}$
 USE TWO WORM GEAR DRIVES AS IN PROBLEM 76.

$$TV_{NOM} = 5500 / 13.5 = 407.4.$$

$$\text{LET } VR_1 = 20. \text{ THEN } VR_2 = 407.4 / 20 = 20.37$$

$$\text{TRY } N_C = 3. N_D = 3(20.37) = 61.11 \Rightarrow \text{USE } 61$$

$$N_A = 3, N_B = 60, N_C = 3, N_D = 61$$

$$\text{FINAL OUTPUT SPEED} = 5500 \times \frac{3}{60} \times \frac{3}{61} = \underline{13.52 \text{ RPM OK}}$$

$$\text{CAN ALSO USE } N_D = 62. \text{ THEN } m_{OUT} = \underline{13.30 \text{ RPM}}$$

$$N_D = 60. \text{ THEN } m_{OUT} = \underline{13.75 \text{ RPM}}$$

CHAPTER 9 SPUR GEAR DESIGN

Forces on Spur Gear Teeth

- 1 GIVEN: $\phi = 20^\circ$, $P = 7.5 \text{ kP}$, $M_P = 1750 \text{ RPM}$, $N_P = 20$, $N_G = 72$, $P_d = 12$
- a) $m_G = M_P \cdot \frac{N_P}{N_G} = 1750 \text{ RPM} \cdot \frac{20}{72} = \underline{486.1 \text{ RPM}}$
- b) $VR = m_G = N_G / N_P = 72 / 20 = \underline{3.600}$
- c) $D_P = N_P / P_d = 20 / 12 = \underline{1.667 \text{ in}}$; $D_G = N_G / P_d = 72 / 12 = \underline{6.000 \text{ in}}$
- d) $C = \frac{N_P + N_G}{2 P_d} = \frac{20 + 72}{2(12)} = \underline{3.833 \text{ in}}$
- e) $N_t = \pi D_P M_P / 12 = \pi (1.667)(1750) / 12 = \underline{764 \text{ FT/MIN}}$
- f) $T_P = \frac{63000(P)}{M_P} = \frac{63000(7.5)}{1750} = \underline{270 \text{ LB}\cdot\text{IN}}$
 $T_G = \frac{63000(P)}{m_G} = \frac{63000(7.5)}{486.1} = \underline{972 \text{ LB}\cdot\text{IN}}$
- g) $W_t = \frac{T_P}{D_P/2} = \frac{270 \text{ LB}\cdot\text{IN}}{1.667 \text{ in}/2} = \underline{324 \text{ LB}}$
 OR $W_t = \frac{33000(P)}{N_t} = \frac{33000(7.5)}{764} = \underline{324 \text{ LB}}$
- h) $W_R = W_t \tan \phi = (324 \text{ LB}) \tan 20^\circ = \underline{118 \text{ LB}}$
- i) $W_N = W_t / \cos \phi = 324 \text{ LB} / \cos 20^\circ = \underline{345 \text{ LB}}$

A SIMILAR METHOD IS USED FOR PROBLEMS 2-6.

SPREADSHEET SOLUTIONS ARE SHOWN ON THE FOLLOWING PAGES. THE SOLUTION FOR PROBLEM 1 IS ALSO SHOWN FOR COMPARISON TO THE SOLUTION SHOWN ABOVE.

Forces on Spur Gear Teeth

Problem 1	Pressure angle =	20	degrees
Chapter 9	Power =	7.5	hp
	pinion speed =	1750	rpm
	teeth in pinion =	20	
	teeth in gear =	72	
	diametral pitch =	12	
RESULTS:			
a	Gear speed =	486.1	rpm
b	VR = m_G =	3.600	
c	pinion PD =	1.667	in
	gear PD =	6.000	in
d	center distance = C =	3.833	in
e	pitch line speed =	764	ft/min
f	torque on pinion shaft =	270	lb in
	torque on gear shaft =	972	lb in
g	tangential force =	324	lb
h	radial force =	118	lb
i	normal force =	345	lb

Problem 2	Pressure angle =	20	degrees
Chapter 9	Power =	50	hp
	pinion speed =	1150	rpm
	teeth in pinion =	18	
	teeth in gear =	68	
	diametral pitch =	5	
RESULTS:			
a	Gear speed =	304.4	rpm
b	VR = m_G =	3.778	
c	pinion PD =	3.600	in
	gear PD =	13.600	in
d	center distance = C =	8.600	in
e	pitch line speed =	1084	ft/min
f	torque on pinion shaft =	2739	lb in
	torque on gear shaft =	10348	lb in
g	tangential force =	1522	lb
h	radial force =	554	lb
i	normal force =	1620	lb

Forces on Spur Gear Teeth

Problem 3	Pressure angle =	20	degrees
Chapter 9	Power =	0.75	hp
	pinion speed =	3450	rpm
	teeth in pinion =	24	
	teeth in gear =	110	
	diametral pitch =	24	
RESULTS:			
a	Gear speed =	752.7	rpm
b	$VR = m_G =$	4.583	
c	pinion PD =	1.000	in
	gear PD =	4.583	in
d	center distance = $C =$	2.792	in
e	pitch line speed =	903	ft/min
f	torque on pinion shaft =	13.70	lb in
	torque on gear shaft =	62.77	lb in
g	tangential force =	27.40	lb
h	radial force =	9.97	lb
i	normal force =	29.16	lb

Problem 4	Pressure angle =	25	degrees
Chapter 9	Power =	7.5	hp
	pinion speed =	1750	rpm
	teeth in pinion =	20	
	teeth in gear =	72	
	diametral pitch =	12	
RESULTS:			
a	Gear speed =	486.1	rpm
b	$VR = m_G =$	3.600	
c	pinion PD =	1.667	in
	gear PD =	6.000	in
d	center distance = $C =$	3.833	in
e	pitch line speed =	764	ft/min
f	torque on pinion shaft =	270	lb in
	torque on gear shaft =	972	lb in
g	tangential force =	324	lb
h	radial force =	151	lb
i	normal force =	358	lb

Forces on Spur Gear Teeth

Problem 5	Pressure angle =	25	degrees
Chapter 9	Power =	50	hp
	pinion speed =	1150	rpm
	teeth in pinion =	18	
	teeth in gear =	68	
	diametral pitch =	5	
RESULTS:			
a	Gear speed =	304.4	rpm
b	VR = m_G =	3.778	
c	pinion PD =	3.600	in
	gear PD =	13.600	in
d	center distance = C =	8.600	in
e	pitch line speed =	1084	ft/min
f	torque on pinion shaft =	2739	lb in
	torque on gear shaft =	10348	lb in
g	tangential force =	1522	lb
h	radial force =	710	lb
i	normal force =	1680	lb

Problem 6	Pressure angle =	25	degrees
Chapter 9	Power =	0.75	hp
	pinion speed =	3450	rpm
	teeth in pinion =	24	
	teeth in gear =	110	
	diametral pitch =	24	
RESULTS:			
a	Gear speed =	752.7	rpm
b	VR = m_G =	4.583	
c	pinion PD =	1.000	in
	gear PD =	4.583	in
d	center distance = C =	2.792	in
e	pitch line speed =	903	ft/min
f	torque on pinion shaft =	13.70	lb in
	torque on gear shaft =	62.77	lb in
g	tangential force =	27.40	lb
h	radial force =	12.78	lb
i	normal force =	30.24	lb

Gear Manufacture and Quality

7. See Section 9-4. Form milling, shaping, hobbing, grinding.

For Problems 8-16, refer to Section 9-5 and Table 9-3 for recommended quality numbers in the A_v system according to AGMA Standard 2015. Grain harvester: $A_v = 10$.

8. Grain harvester: $A_v = 10$.
9. Printing press: $A_v = 7$.
10. Auto transmission: $A_v = 6$.
11. Gyroscope: $A_v = 2$.
12. Analytical quality measurements include *index variation, tooth alignment, tooth profile, root radius, and runout*.
13. AGMA Standard 2015 is currently used. See Table 9-2 for the range of quality numbers in this system and the comparisons with prior systems.

For Problems 14-16, for precision machinery, use the recommendations for machine tool drives in the lower part of Table 9-3. The choice of quality number is based on the pitch line speed of the gears.

14. (From Problem 1). Pitch line speed = 764 ft/min Use $A_v = 10$.
15. (From Problem 2). Pitch line speed = 1084 ft/min Use $A_v = 8$.
16. (From Problem 3). Pitch line speed = 903 ft/min Use $A_v = 8$.

Gear Materials

Answers for Problems 17 – 25 are found in Sections 9-6 and 9-7. Only brief statements are given here.

17. Bending stresses are created by the tangential force on the gear teeth acting in a manner similar to that on a cantilever. The maximum bending stress occurs in the root of the tooth where it blends with the involute tooth form. High levels of contact stress, called Hertz stress, occur in the face of the teeth near the pitch line as forces are exerted between the pinion and the gear teeth. The probable mode of failure is pitting of the tooth surface.
18. AGMA standards give allowable bending stress numbers and allowable contact stress numbers related to the hardness of the material of the teeth. See Figures 9-11 and 9-12.
19. Gear steels are typically medium carbon plain or alloy steels that are heat treated by through-hardening using a quenching and tempering process. For examples, see Table 9-4, Section 9-7.
20. The AGMA recommends hardness values from HB 180 to HB 400. See Figures 9-11 and 9-12.
21. Grade 1 steel is typical commercial quality and is recommended for use in this book. Grades 2 and 3 require progressively more stringent quality controls on the alloy content and cleanliness of the materials. Cost increases dramatically for the higher grades. See AGMA Standard 2004-C08 or the latest revision.
22. Grades 2 and 3 may be specified for high-speed aerospace applications, turbine engine driven systems, ship propulsion drives, and high-capacity industrial drives such as those in steel rolling mills.

23. Case hardening by flame hardening, induction hardening, and carburizing are three processes that produce harder surfaces than typical through-hardening.
24. See AGMA Standard 2001-D04 or the latest revision.
25. AGMA Standard 2001-D04 provides data for gray cast iron, ductile iron, and bronze. Table 9-6.
26. From Figures 9-11 and 9-12:
 - a. Grade 1; 200 HB: $s_{at} = 28.26$ ksi; $s_{ac} = 93.50$ ksi – U.S.: $s_{at} = 194.9$ MPa; $s_{ac} = 644.6$ MPa – SI
 - b. Grade 1; 300 HB: $s_{at} = 36.0$ ksi; $s_{ac} = 125.7$ ksi – U.S.: $s_{at} = 248.1$ MPa; $s_{ac} = 866.6$ MPa – SI
 - c. Grade 1; 400 HB: $s_{at} = 43.72$ ksi; $s_{ac} = 157.9$ ksi – U.S.: $s_{at} = 301.5$ MPa; $s_{ac} = 1088.6$ MPa – SI
 - d. Using HB > 400 is not recommended.
 - e. Grade 2; 200 HB: $s_{at} = 36.80$ ksi; $s_{ac} = 104.1$ ksi – U.S.: $s_{at} = 253.7$ MPa; $s_{ac} = 718.5$ MPa – SI
 - f. Grade 2; 300 HB: $s_{at} = 47.0$ ksi; $s_{ac} = 139.0$ ksi – U.S.: $s_{at} = 324.0$ MPa; $s_{ac} = 959.5$ MPa – SI
 - g. Grade 2; 400 HB: $s_{at} = 57.20$ ksi; $s_{ac} = 173.9$ ksi – U.S.: $s_{at} = 394.3$ MPa; $s_{ac} = 1200.5$ MPa – SI
27. From Figure 9-11: Grade 1: 300 HB. Grade 2: 192 HB
28. From Table 9-5: Case hardening by carburizing produces 55-64 HRC
29. From Appendix 5: SAE 1020, 4118, 8620, and others
30. From Table 9-5: Flame or induction hardening produces 50-54 HRC with materials having high hardenability
31. SAE 4140, 4340, 6150. All have good hardenability
32. ASTM A536, Grade 80-55-06 has a minimum hardness of 179 HB.
33.
 - a. $s_{at} = 45.0$ ksi; $s_{ac} = 170.0$ ksi – U.S.: $s_{at} = 310$ MPa; $s_{ac} = 1172$ MPa – SI [Table 9-5]
 - b. $s_{at} = 45.0$ ksi; $s_{ac} = 175.0$ ksi – U.S.: $s_{at} = 310$ MPa; $s_{ac} = 1207$ MPa – SI [Table 9-5]
 - c. $s_{at} = 55.0$ ksi; $s_{ac} = 180.0$ ksi – U.S.: $s_{at} = 379$ MPa; $s_{ac} = 1241$ MPa – SI [Table 9-5]
 - d. Not listed
 - e. $s_{at} = 55.0$ ksi; $s_{ac} = 180.0$ ksi – U.S.: $s_{at} = 379$ MPa; $s_{ac} = 1241$ MPa – SI [Table 9-5]
 - f. $s_{at} = 5.00$ ksi; $s_{ac} = 50.0$ ksi – U.S.: $s_{at} = 35.0$ MPa; $s_{ac} = 345$ MPa – SI [Table 9-6]
 - g. $s_{at} = 13.0$ ksi; $s_{ac} = 75.0$ ksi – U.S.: $s_{at} = 90.0$ MPa; $s_{ac} = 517$ MPa – SI [Table 9-6]
 - h. $s_{at} = 27.0$ ksi; $s_{ac} = 92.0$ ksi – U.S.: $s_{at} = 186$ MPa; $s_{ac} = 634$ MPa – SI [Table 9-6]
 - i. $s_{at} = 5.70$ ksi; $s_{ac} = 30.0$ ksi – U.S.: $s_{at} = 39.0$ MPa; $s_{ac} = 207$ MPa – SI [Table 9-6]
 - j. $s_{at} = 23.6$ ksi; $s_{ac} = 65.0$ ksi – U.S.: $s_{at} = 163$ MPa; $s_{ac} = 448$ MPa – SI [Table 9-6]
 - k. $s_{at} = 12.0$ ksi; s_{ac} not listed: $s_{at} = 83.0$ MPa; s_{ac} not listed [Table 9-14]
 - l. $s_{at} = 9.0$ ksi; s_{ac} not listed: $s_{at} = 62.0$ MPa; s_{ac} not listed [Table 9-14]
34. Depth = 0.027 in [Figure 9-13.]
35. Depth = 0.90 mm. [Figure 9-13.]

DESIGN OF SPUR GEARS

APPLICATION: Problems 36, 42, 48, 54		Factors in Design Analysis:	
Industrial conveyor driven by an electric motor		Alignment Factor, $K_m = 1.0 + C_{pf} + C_{ma}$	If $F < 1.0$ If $F > 1.0$
Initial Input Data:		Pinion Proportion Factor, $C_{pf} =$	0.058 0.061 [0.50 < F/D _p < 2.00]
Overload Factor: $K_o =$	1.50 Table 9-7	Enter: $C_{pf} =$	0.061 Figure 9-16
Transmitted Power: $P =$	10 hp	Type of gearing:	Open Commer. Precision Ex. Prec.
Design Power $P_{des} =$	15 hp	Mesh Alignment Factor, $C_{ma} =$	0.268 0.147 0.083 0.051
Diametral Pitch: $P_d =$	12 Fig. 9-24	Enter: $C_{ma} =$	0.147 Figure 9-17
Input Speed: $n_P =$	1750 rpm	Alignment Factor: $K_m =$	1.21 [Computed]
Number of Pinion Teeth: $N_P =$	18	Size Factor: $K_s =$	1.00 Table 9-8: Use 1.00 if $P_d \geq 5$
Desired Output Speed: $n_G =$	370 rpm	Pinion Rim Thickness Factor: $K_{RP} =$	1.00 Fig. 9-18: Use 1.00 if solid blank
Computed number of gear teeth:	85.1	Gear Rim Thickness Factor: $K_{RG} =$	1.00 Fig. 9-18: Use 1.00 if solid blank
Enter: Chosen No. of Gear Teeth: $N_G =$	85	Service Factor: $SF =$	1.00 Use 1.00 if no unusual conditions
Computed data:		Reliability Factor: $K_R =$	1.00 Table 9-11 Use 1.00 for $R = .99$
Actual Output Speed: $n_G =$	370.6 rpm	Enter: Design Life: 20000 hours	See Table 9-12
Gear Ratio: $m_G =$	4.72	Pinion - Number of load cycles: $N_P =$	2.1E+09
Pitch Diameter - Pinion: $D_P =$	1.500 in	Gear - Number of load cycles: $N_G =$	4.4E+08
Pitch Diameter - Gear: $D_G =$	7.083 in	Bending Stress Cycle Factor: $Y_{NP} =$	0.93 Fig. 9-22
Center Distance: $C =$	4.292 in	Bending Stress Cycle Factor: $Y_{NG} =$	0.95 Fig. 9-22
Pitch Line Speed: $v_t =$	687 ft/min	Pitting Stress Cycle Factor: $Z_{NP} =$	0.88 Fig. 9-23
Transmitted Load: $W_t =$	480 lb	Pitting Stress Cycle Factor: $Z_{NG} =$	0.92 Fig. 9-23
Secondary Input Data:		Stress Analysis: Bending	
Face Width Guidelines (in):	Min Nom Max	Pinion: Required $s_{at} =$	37,906 psi See Fig. 9-11 or
Enter: Face Width: $F =$	0.667 1.000 1.333	Gear: Required $s_{at} =$	28,963 psi Table 9-5
Ratio: Face width/pinion diameter: $F/D_P =$	0.83	Stress Analysis: Pitting	
Recommended range of ratio: $0.50 < F/D_P < 2.00$		Pinion: Required $s_{ec} =$	199,099 psi See Fig. 9-12 or
Enter: Elastic Coefficient: $C_p =$	2300 Table 9-10	Gear: Required $s_{ec} =$	190,443 psi Table 9-5
Enter: Quality Number: $A_v =$	11 Table 9-3	Required hardness of pinion HB: 528 Equations in Fig. 9-12-Grade 1	
Dynamic Factor: $K_v =$	1.35 Table 9-9	Required hardness of gear HB: 501 Equations in Fig. 9-12-Grade 1	
[Factors for computing K_v]: $B =$	0.826 $C = 59.75$	Specify materials, alloy and heat treatment, for most severe requirement.	
Reference: $N_P = 18$ $N_G = 85$		One possible material specification:	
Bending Geometry Factor-Pinion: $J_P =$	0.320 Fig. 9-15	Requires Grade 2 carburized. Suggest redesign to lower stress levels.	
Bending Geometry Factor-Gear: $J_G =$	0.410 Fig. 9-15		
Reference: $m_G = 4.72$			
Enter: Pitting Geometry Factor: $I =$	0.108 Fig. 9-21		

Computed bending stress number, $s_t =$	35253 psi	Pinion
Computed bending stress number, $s_t =$	27514 psi	Gear
Computed contact stress number, $s_c =$	175,207 psi	Pinion
Computed contact stress number, $s_c =$	175,207 psi	Gear

Ans. Problem:	36
Ans. Problem:	36
Ans. Problem:	48
Ans. Problem:	48

DESIGN OF SPUR GEARS

APPLICATION: Problems 37, 43, 49, 55		Factors in Design Analysis:	
Cement kiln driven by an electric motor		Alignment Factor, $K_m = 1.0 + C_{pf} + C_{ma}$	If $F < 1.0$ If $F > 1.0$
Initial Input Data:		Pinion Proportion Factor, $C_{pf} =$	0.043 0.058 [0.50 < F/D_p < 2.00]
Overload Factor: $K_o =$	1.75 Table 9-7	Enter: $C_{pf} =$	0.058 Figure 9-16
Transmitted Power: $P =$	40 hp	Type of gearing: Open	Commer. Precision Ex. Prec.
Design Power $P_{des} =$	70 hp	Mesh Alignment Factor, $C_{ma} =$	0.284 0.162 0.096 0.061
Diametral Pitch: $P_d =$	6 Fig. 9-24	Enter: $C_{ma} =$	0.162 Figure 9-17
Input Speed: $n_P =$	1150 rpm	Alignment Factor: $K_m =$	1.22 [Computed]
Number of Pinion Teeth: $N_P =$	20	Size Factor: $K_s =$	1.00 Table 9-8: Use 1.00 if $P_d \geq 5$
Desired Output Speed: $n_G =$	479 rpm	Pinion Rim Thickness Factor: $K_{sp} =$	1.00 Fig. 9-18: Use 1.00 if solid blank
Computed number of gear teeth: 48.0		Gear Rim Thickness Factor: $K_{sg} =$	1.00 Fig. 9-18: Use 1.00 if solid blank
Enter: Chosen No. of Gear Teeth: $N_G =$	48	Service Factor: $SF =$	1.00 Use 1.00 if no unusual conditions
Computed data:		Reliability Factor: $K_R =$	1.00 Table 9-11 Use 1.00 for $R = .99$
Actual Output Speed: $n_G =$	479.2 rpm	Enter: Design Life: 8000 hours	See Table 9-12
Gear Ratio: $m_G =$	2.40	Pinion - Number of load cycles: $N_P =$	5.5E+08
Pitch Diameter - Pinion: $D_P =$	3.333 in	Gear - Number of load cycles: $N_G =$	2.3E+08
Pitch Diameter - Gear: $D_G =$	8.000 in	Bending Stress Cycle Factor: $Y_{NP} =$	0.95 Fig. 9-22
Center Distance: $C =$	5.667 in	Bending Stress Cycle Factor: $Y_{NG} =$	0.96 Fig. 9-22
Pitch Line Speed: $v_t =$	1004 ft/min	Pitting Stress Cycle Factor: $Z_{NP} =$	0.91 Fig. 9-23
Transmitted Load: $W_t =$	1315 lb	Pitting Stress Cycle Factor: $Z_{NG} =$	0.93 Fig. 9-23
Secondary Input Data:		Stress Analysis: Bending	
Face Width Guidelines (in): 1.333 2.000 2.667	Min Nom Max	Pinion: Required $s_{at} =$	34,466 psi See Fig. 9-11 or
Enter: Face Width: $F =$	2.250 in	Gear: Required $s_{at} =$	28,063 psi Table 9-5
Ratio: Face width/pinion diameter: $F/D_P =$	0.68	Stress Analysis: Pitting	
Recommended range of ratio: $0.50 < F/D_P < 2.00$		Pinion: Required $s_{ec} =$	189,152 psi See Fig. 9-12 or
Enter: Elastic Coefficient: $C_p =$	2300 Table 9-10	Gear: Required $s_{ec} =$	185,084 psi Table 9-5
Enter: Quality Number: $A_v =$	11 Table 9-3	Required hardness of pinion HB: 497 Equations in Fig. 9-12-Grade 1	
Dynamic Factor: $K_v =$	1.42 Table 9-9	Required hardness of gear HB: 484 Equations in Fig. 9-12-Grade 1	
[Factors for computing K_v]: $B =$	0.828 $C =$	Specify materials, alloy and heat treatment, for most severe requirement.	
Reference: $N_P =$	20 $N_G =$	One possible material specification:	
Bending Geometry Factor-Pinion: $J_P =$	0.325 Fig. 9-15	Requires Grade 2 flame or induction harden. Suggest redesign to lower stress levels.	
Bending Geometry Factor-Gear: $J_G =$	0.395 Fig. 9-15		
Reference: $m_G =$	2.40		
Enter: Pitting Geometry Factor: $I =$	0.095 Fig. 9-21		

Computed bending stress number, $s_t =$	32743 psi	Pinion
Computed bending stress number, $s_t =$	26940 psi	Gear
Computed contact stress number, $s_c =$	172,128 psi	Pinion
Computed contact stress number, $s_c =$	172,128 psi	Gear

Ans. Problem:	37
Ans. Problem:	37
Ans. Problem:	49
Ans. Problem:	49

DESIGN OF SPUR GEARS	
APPLICATION: [Problems 38, 44, 50, 56] Small machine tool driven by an electric motor	
Initial Input Data: Overload Factor: $K_o = 1.50$ Table 9-7 Transmitted Power: $P = 0.5$ hp Design Power $P_{des} = 0.75$ hp Diametral Pitch: $P_d = 32$ Fig. 9-24 Input Speed: $n_P = 3450$ rpm Number of Pinion Teeth: $N_P = 24$ Desired Output Speed: $n_G = 690$ rpm Computed number of gear teeth: 120.0 Enter: Chosen No. of Gear Teeth: $N_G = 120$	
Computed data: Actual Output Speed: $n_G = 690.0$ rpm Gear Ratio: $m_G = 5.00$ Pitch Diameter - Pinion: $D_P = 0.750$ in Pitch Diameter - Gear: $D_G = 3.750$ in Center Distance: $C = 2.250$ in Pitch Line Speed: $v_t = 677$ ft/min Transmitted Load: $W_t = 24$ lb	
Secondary Input Data: Face Width Guidelines (in): Min 0.250 Nom 0.375 Max 0.500 Enter: Face Width: $F = 0.500$ in Ratio: Face width/pinion diameter: $F/D_P = 0.67$ Recommended range of ratio: $0.50 < F/D_P < 2.00$	
Enter: Elastic Coefficient: $C_p = 2300$ Table 9-10 Enter: Quality Number: $A_v = 7$ Table 9-3 Dynamic Factor: $K_v = 1.11$ Table 9-9 [Factors for computing K_v] $B = 0.397$ $C = 83.77$	
Reference: $N_P = 24$ $N_G = 120$ Bending Geometry Factor-Pinion: $J_P = 0.360$ Fig. 9-15 Bending Geometry Factor-Gear: $J_G = 0.440$ Fig. 9-15 Reference: $m_G = 5.00$	
Enter: Pitting Geometry Factor: $I = 0.118$ Fig. 9-21	
Ans. Problem: 38 Ans. Problem: 38 Ans. Problem: 50 Ans. Problem: 50	

Factors in Design Analysis:			
Alignment Factor, $K_m = 1.0 + C_{pt} + C_{ma}$	If $F < 1.0$	If $F > 1.0$	
Pinion Proportion Factor, $C_{pt} = 0.042$	0.042	0.035	$[0.50 < F/D_P < 2.00]$
Enter: $C_{pt} = 0.042$ Figure 9-16			
Type of gearing: Open	Commer.	Precision	Ex. Prec.
Mesh Alignment Factor, $C_{ma} = 0.255$	0.255	0.135	0.074
Enter: $C_{ma} = 0.074$ Figure 9-17			
Alignment Factor: $K_m = 1.12$	[Computed]		
Size Factor: $K_s = 1.00$ Table 9-8: Use 1.00 if $P_d \geq 5$			
Pinion Rim Thickness Factor: $K_{RP} = 1.00$	Fig. 9-18: Use 1.00 if solid blank		
Gear Rim Thickness Factor: $K_{RG} = 1.00$	Fig. 9-18: Use 1.00 if solid blank		
Service Factor: $SF = 1.25$	Use 1.00 if no unusual conditions		
Reliability Factor: $K_R = 1.50$	Table 9-11 Use 1.00 for $R = .99$		
Enter: Design Life: 12000 hours See Table 9-12			
Pinion - Number of load cycles: $N_P = 2.5E+08$	Guidelines: Y_N, Z_N		
Gear - Number of load cycles: $N_G = 5.0E+08$	10^7 cycles	$> 10^7$	$< 10^7$
Bending Stress Cycle Factor: $Y_{NP} = 0.92$	1.00	0.92	Fig. 9-22
Bending Stress Cycle Factor: $Y_{NG} = 0.95$	1.00	0.95	Fig. 9-22
Pitting Stress Cycle Factor: $Z_{NP} = 0.88$	1.00	0.88	Fig. 9-23
Pitting Stress Cycle Factor: $Z_{NG} = 0.91$	1.00	0.91	Fig. 9-23
Stress Analysis: Bending			
Pinion: Required $s_{at} = 16,448$ psi			See Fig. 9-11 or Table 9-5
Gear: Required $s_{at} = 13,033$ psi			
Stress Analysis: Pitting			
Pinion: Required $s_{ac} = 156,968$ psi			See Fig. 9-12 or Table 9-5
Gear: Required $s_{ac} = 151,791$ psi			
Required hardness of pinion HB: 397 Equations in Fig. 9-12-Grade 1			
Required hardness of gear HB: 381 Equations in Fig. 9-12-Grade 1			
Specify materials, alloy and heat treatment, for most severe requirement.			
One possible material specification:			
SAE 4140 OQT 800, 429 HB, 14% elongation, good ductility			
SAE 4140 OQT 900, 429 HB, 14% elongation, good ductility			

Computed bending stress number, $s_t = 8071$ psi	Pinion
Computed bending stress number, $s_t = 6603$ psi	Gear
Computed contact stress number, $s_c = 73,669$ psi	Pinion
Computed contact stress number, $s_c = 73,669$ psi	Gear

DESIGN OF SPUR GEARS									
APPLICATION: Problems 39, 45, 51, 57 Aircraft actuator driven by a universal electric motor									
Initial Input Data:									
Overload Factor:	$K_o = 1.50$	Table 9-7							
Transmitted Power:	$P = 15$	hp							
Design Power P_{des} :	$P_{des} = 22.5$	hp							
Diametral Pitch:	$P_d = 10$	Fig. 9-24							
Input Speed:	$n_P = 6500$	rpm							
Number of Pinion Teeth:	$N_P = 30$								
Desired Output Speed:	$n_G = 2216$	rpm							
Computed number of gear teeth:	88.0								
Enter: Chosen No. of Gear Teeth:	$N_G = 88$								
Computed data:									
Actual Output Speed:	$n_G = 2215.9$	rpm							
Gear Ratio:	$m_G = 2.93$								
Pitch Diameter - Pinion:	$D_P = 3.000$	in							
Pitch Diameter - Gear:	$D_G = 8.800$	in							
Center Distance:	$C = 5.900$	in							
Pitch Line Speed:	$v_t = 5105$	ft/min							
Transmitted Load:	$W_t = 97$	lb							
Secondary Input Data:									
Face Width Guidelines (in):	Min	Nom	Max						
Enter: Face Width:	$F = 1.500$	in							
Ratio: Face width/pinion diameter:	$F/D_P = 0.50$								
Recommended range of ratio:	$0.50 < F/D_P < 2.00$								
Enter: Elastic Coefficient:	$C_p = 2300$	Table 9-10							
Enter: Quality Number:	$A_v = 5$	Table 9-3							
Dynamic Factor:	$K_v = 1.00$	Table 9-9							
[Factors for computing K_v]	$B = 0.000$	$C = 106.00$							
Reference:	$N_P = 30$	$N_G = 88$							
Bending Geometry Factor-Pinion:	$J_P = 0.460$	Fig. 9-15							
Bending Geometry Factor-Gear:	$J_G = 0.530$	Fig. 9-15							
Reference:	$m_G = 2.93$								
Enter: Pitting Geometry Factor:	$I = 0.130$	Fig. 9-21							
Factors in Design Analysis:									
Alignment Factor, $K_m = 1.0 + C_{pt} + C_{ma}$	If $F < 1.0$	If $F > 1.0$							
Pinion Proportion Factor, C_{pt}	$C_{pt} = 0.025$	$C_{pt} = 0.031$	$[0.50 < F/D_P < 2.00]$						
Enter: $C_{pt} = 0.031$	Figure 9-16								
Type of gearing:	Open	Commer.	Precision	Ex. Prec.					
Mesh Alignment Factor, C_{ma}	$C_{ma} = 0.272$	$C_{ma} = 0.150$	0.086	0.053					
Enter: $C_{ma} = 0.053$	Figure 9-17								
Alignment Factor: K_m	$K_m = 1.08$	[Computed]							
Size Factor: K_s	$K_s = 1.00$	Table 9-8: Use 1.00 if $P_d \geq 5$							
Pinion Rim Thickness Factor: K_{RP}	$K_{RP} = 1.00$	Fig. 9-18: Use 1.00 if solid blank							
Gear Rim Thickness Factor: K_{RG}	$K_{RG} = 1.00$	Fig. 9-18: Use 1.00 if solid blank							
Service Factor: SF	$SF = 1.00$	Use 1.00 if no unusual conditions							
Reliability Factor: K_R	$K_R = 1.50$	Table 9-11 Use 1.00 for $R = .99$							
Enter: Design Life:	4000	hours							
Pinion - Number of load cycles: N_P	$N_P = 1.6E+09$	Guidelines: Y_N, Z_N							
Gear - Number of load cycles: N_G	$N_G = 5.3E+08$	10^7 cycles	$> 10^7$	$< 10^7$					
Bending Stress Cycle Factor: Y_{NP}	$Y_{NP} = 0.93$	1.00	0.93	Fig. 9-22					
Bending Stress Cycle Factor: Y_{NG}	$Y_{NG} = 0.95$	1.00	0.95	Fig. 9-22					
Pitting Stress Cycle Factor: Z_{NP}	$Z_{NP} = 0.89$	1.00	0.89	Fig. 9-23					
Pitting Stress Cycle Factor: Z_{NG}	$Z_{NG} = 0.91$	1.00	0.91	Fig. 9-23					
Stress Analysis: Bending									
Pinion: Required s_{at}	$s_{at} = 3,685$	psi	See Fig. 9-11 or Table 9-5						
Gear: Required s_{at}	$s_{at} = 3,131$	psi							
Stress Analysis: Pitting									
Pinion: Required s_{ec}	$s_{ec} = 63,637$	psi	See Fig. 9-12 or Table 9-5						
Gear: Required s_{ec}	$s_{ec} = 62,239$	psi							
Required hardness of pinion HB:	107	Equations in Fig. 9-12-Grade 1							
Required hardness of gear HB:	103	Equations in Fig. 9-12-Grade 1							
Specify materials, alloy and heat treatment, for most severe requirement.									
One possible material specification:									
Stresses are quite low for steel gears. Suggest redesign.									
Computed bending stress number, s_t									
Pinion	2285	psi							
Gear	1983	psi							
Computed contact stress number, s_c									
Pinion	37,758	psi							
Gear	37,758	psi							

DESIGN OF SPUR GEARS									
APPLICATION: [Problems 40, 46, 52, 58 Portable industrial water pump driven by a gasoline engine]									
Initial Input Data:									
Overload Factor: $K_o = 1.70$ Table 9-7 Transmitted Power: $P = 125$ hp Design Power $P_{des} = 212.5$ hp Diametral Pitch: $P_d = 4$ Fig. 9-24 Input Speed: $n_P = 2500$ rpm Number of Pinion Teeth: $N_P = 32$ Desired Output Speed: $n_G = 1050$ rpm Computed number of gear teeth: 76.2 Enter: Chosen No. of Gear Teeth: $N_G = 76$									
Computed data:									
Actual Output Speed: $n_G = 1052.6$ rpm Gear Ratio: $m_G = 2.38$ Pitch Diameter - Pinion: $D_P = 8.000$ in Pitch Diameter - Gear: $D_G = 19.000$ in Center Distance: $C = 13.500$ in Pitch Line Speed: $v_t = 5236$ ft/min Transmitted Load: $W_t = 788$ lb									
Secondary Input Data:									
Face Width Guidelines (in): Min 2.000 Nom 3.000 Max 4.000 Enter: Face Width: $F = 1.500$ in Ratio: Face width/pinion diameter: $F/D_P = 0.50$ Entered Recommended range of ratio: $0.50 < F/D_P < 2.00$									
Enter: Elastic Coefficient: $C_p = 2300$ Table 9-10 Enter: Quality Number: $A_v = 9$ Table 9-3 Dynamic Factor: $K_v = 1.56$ Table 9-9 [Factors for computing K_v]: $B = 0.630$ $C = 70.71$									
Reference: $N_P = 32$ $N_G = 76$ Bending Geometry Factor-Pinion: $J_P = 0.465$ Fig. 9-15 Bending Geometry Factor-Gear: $J_G = 0.520$ Fig. 9-15 Reference: $m_G = 2.38$									
Enter: Pitting Geometry Factor: $I = 0.124$ Fig. 9-21									
Ans. Problem: 40 Ans. Problem: 40 Ans. Problem: 52 Ans. Problem: 52									
Factors in Design Analysis:									
Alignment Factor, $K_m = 1.0 + C_{pf} + C_{ma}$ If $F < 1.0$ If $F > 1.0$ Pinion Proportion Factor, $C_{pf} = 0.025$ 0.031 $[0.50 < F/D_P < 2.00]$ Enter: $C_{pf} = 0.031$ Figure 9-16 Type of gearing: Open Commer. Precision Ex. Prec. Mesh Alignment Factor, $C_{ma} = 0.272$ 0.150 0.086 0.053 Enter: $C_{ma} = 0.150$ Figure 9-17 Alignment Factor: $K_m = 1.18$ [Computed]									
Size Factor: $K_s = 1.05$ Table 9-8: Use 1.00 if $P_d \geq 5$ Pinion Rim Thickness Factor: $K_{RP} = 1.00$ Fig. 9-18: Use 1.00 if solid blank Gear Rim Thickness Factor: $K_{RG} = 1.00$ Fig. 9-18: Use 1.00 if solid blank Service Factor: $S_F = 1.00$ Use 1.00 if no unusual conditions Reliability Factor: $K_R = 1.00$ Table 9-11 Use 1.00 for $R = .99$									
Enter: Design Life: 8000 hours See Table 9-12 Pinion - Number of load cycles: $N_P = 1.2E+09$ Guidelines: Y_N, Z_N Gear - Number of load cycles: $N_G = 5.1E+08$ 10 ⁷ cycles $> 10^7$ $< 10^7$ Bending Stress Cycle Factor: $Y_{NP} = 0.93$ 1.00 0.93 Fig. 9-22 Bending Stress Cycle Factor: $Y_{NG} = 0.95$ 1.00 0.95 Fig. 9-22 Pitting Stress Cycle Factor: $Z_{NP} = 0.90$ 1.00 0.90 Fig. 9-23 Pitting Stress Cycle Factor: $Z_{NG} = 0.91$ 1.00 0.91 Fig. 9-23									
Stress Analysis: Bending Pinion: Required $s_{at} = 15,968$ psi See Fig. 9-11 or Gear: Required $s_{at} = 13,979$ psi Table 9-5									
Stress Analysis: Pitting Pinion: Required $s_{hc} = 106,809$ psi See Fig. 9-12 or Gear: Required $s_{hc} = 105,438$ psi Table 9-5									
Required hardness of pinion HB: 241 Equations in Fig. 9-12-Grade 1 Required hardness of gear HB: 237 Equations in Fig. 9-12-Grade 1 Specify materials, alloy and heat treatment, for most severe requirement.									
One possible material specification: SAE 1040 WQT 1000, 269 HB, 22% elongation SAE 1040 WQT 1000, 269 HB, 22% elongation									
Computed bending stress number, $s_t = 14850$ psi Pinion Computed bending stress number, $s_t = 13280$ psi Gear Computed contact stress number, $s_c = 95,948$ psi Pinion Computed contact stress number, $s_c = 95,948$ psi Gear									

DESIGN OF SPUR GEARS									
APPLICATION: Problems 41, 47, 53, 59 Lawn and garden tractor with fluid motor drive									
Initial Input Data:									
Overload Factor: $K_o = 1.75$ Table 9-7 Transmitted Power: $P = 2.5$ hp Design Power $P_{des} = 4.375$ hp Diametral Pitch: $P_d = 10$ Fig. 9-24 Input Speed: $n_P = 680$ rpm Number of Pinion Teeth: $N_P = 24$ Desired Output Speed: $n_G = 263$ rpm Computed number of gear teeth: 62.1 Enter: Chosen No. of Gear Teeth: $N_G = 62$									
Computed data:									
Actual Output Speed: $n_G = 263.2$ rpm Gear Ratio: $m_G = 2.58$ Pitch Diameter - Pinion: $D_P = 2.400$ in Pitch Diameter - Gear: $D_G = 6.200$ in Center Distance: $C = 4.300$ in Pitch Line Speed: $v_t = 427$ ft/min Transmitted Load: $W_t = 193$ lb									
Secondary Input Data:									
Face Width Guidelines (in): Min 0.800 Nom 1.200 Max 1.600 Enter: Face Width: $F = 1.250$ in Ratio: Face width/pinion diameter: $F/D_P = 0.52$ Recommended range of ratio: $0.50 < F/D_P < 2.00$ Enter: Elastic Coef. (Ductile Iron) $C_p = 2100$ Table 9-10 Enter: Quality Number: $A_v = 11$ Table 9-3 Dynamic Factor: $K_v = 1.28$ Table 9-9 [Factors for computing K_v]: $B = 0.826$ $C = 59.75$ Reference: $N_P = 24$ $N_G = 62$ Bending Geometry Factor-Pinion: $J_P = 0.430$ Fig. 9-15 Bending Geometry Factor-Gear: $J_G = 0.500$ Fig. 9-15 Reference: $m_G = 2.58$ Enter: Pitting Geometry Factor: $I = 0.122$ Fig. 9-21									
Ans. Problem: 41 Ans. Problem: 41 Ans. Problem: 53 Ans. Problem: 53									
Factors in Design Analysis:									
Alignment Factor, $K_m = 1.0 + C_{pf} + C_{ma}$ If $F < 1.0$ 0.027 If $F > 1.0$ 0.030 $[0.50 < F/D_P < 2.00]$ Pinion Proportion Factor, $C_{pf} = 0.030$ Figure 9-16 Enter: $C_{pf} = 0.030$ Type of gearing: Open Commer. Precision Ex. Prec. Mesh Alignment Factor, $C_{ma} = 0.268$ 0.147 0.083 0.051 Enter: $C_{ma} = 0.147$ Figure 9-17 Alignment Factor: $K_m = 1.18$ [Computed]									
Size Factor: $K_s = 1.00$ Table 9-8: Use 1.00 if $P_d \geq 5$ Pinion Rim Thickness Factor: $K_{RP} = 1.00$ Fig. 9-18: Use 1.00 if solid blank Gear Rim Thickness Factor: $K_{RG} = 1.00$ Fig. 9-18: Use 1.00 if solid blank Service Factor: $SF = 1.25$ Use 1.00 if no unusual conditions Reliability Factor: $K_R = 0.85$ Table 9-11 Use 1.00 for $R = .99$									
Enter: Design Life: 2000 hours See Table 9-12 Pinion - Number of load cycles: $N_P = 8.2E+07$ Guidelines: Y_N, Z_N Gear - Number of load cycles: $N_G = 3.2E+07$ 10 ⁷ cycles $> 10^7$ $< 10^7$									
Bending Stress Cycle Factor: $Y_{NP} = 0.98$ 1.00 0.98 Fig. 9-22 Bending Stress Cycle Factor: $Y_{NG} = 1.00$ 1.00 1.00 Fig. 9-22 Pitting Stress Cycle Factor: $Z_{NP} = 0.95$ 1.00 0.95 Fig. 9-23 Pitting Stress Cycle Factor: $Z_{NG} = 0.97$ 1.00 0.97 Fig. 9-23									
Stress Analysis: Bending Pinion: Required $s_{at} = 10,254$ psi See Fig. 9-11 or Gear: Required $s_{at} = 8,642$ psi Table 9-5									
Stress Analysis: Pitting Pinion: Required $s_{ec} = 87,531$ psi See Fig. 9-12 or Gear: Required $s_{ec} = 85,727$ psi Table 9-5									
Required hardness of pinion HB: 181 Equations in Fig. 9-12-Grade 1 Required hardness of gear HB: 176 Equations in Fig. 9-12-Grade 1 Specify materials, alloy and heat treatment, for most severe requirement.									
One possible material specification: Pinion: Ductile Iron 100-70-03 Q&T; $s_{at} = 27,000$ psi; $s_{ec} = 92,000$ psi Gear: Ductile Iron 100-70-03 Q&T; $s_{at} = 27,000$ psi; $s_{ec} = 92,000$ psi									
Computed bending stress number, s_t = 9458 psi Pinion Computed bending stress number, s_t = 8134 psi Gear Computed contact stress number, s_c = 78,263 psi Pinion Computed contact stress number, s_c = 78,263 psi Gear									

DESIGN OF SPUR GEARS

APPLICATION: Problem 60		Factors in Design Analysis:			
Reciprocating compressor driven by an electric motor		Alignment Factor, $K_m = 1.0 + C_{pf} + C_{ma}$		If $F < 1.0$ If $F > 1.0$	
Initial Input Data:		Pinion Proportion Factor, $C_{pf} =$		0.044 0.048 [0.50 < $F/D_p < 2.00$]	
Overload Factor: $K_o =$ 1.50 Table 9-7		Enter: $C_{pf} =$ 0.048		Figure 9-16	
Transmitted Power: $P =$ 5 hp		Type of gearing: Open		Commer. Precision Ex. Prec.	
Design Power $P_{des} =$ 7.5 hp		Mesh Alignment Factor, $C_{ma} =$ 0.268		0.147 0.083 0.051	
Diametral Pitch: $P_d =$ 10 Fig. 9-24		Enter: $C_{ma} =$ 0.147		Figure 9-17	
Input Speed: $n_p =$ 1200 rpm		Alignment Factor: $K_m =$ 1.20		[Computed]	
Number of Pinion Teeth: $N_p =$ 18		Size Factor: $K_s =$ 1.00		Table 9-8: Use 1.00 if $P_d \geq 5$	
Desired Output Speed: $n_g =$ 387.5 rpm		Pinion Rim Thickness Factor: $K_{sP} =$ 1.00		Fig. 9-18: Use 1.00 if solid blank	
Computed number of gear teeth: 55.7		Gear Rim Thickness Factor: $K_{sG} =$ 1.00		Fig. 9-18: Use 1.00 if solid blank	
Enter: Chosen No. of Gear Teeth: $N_g =$ 56		Service Factor: $SF =$ 1.00		Use 1.00 if no unusual conditions	
Computed data:		Reliability Factor: $K_R =$ 1.00		Table 9-11 Use 1.00 for $R = .99$	
Actual Output Speed: $n_g =$ 385.7 rpm		Enter: Design Life: 20000 hours		See Table 9-12	
Gear Ratio: $m_g =$ 3.11		Pinion - Number of load cycles: $N_p =$ 1.4E+09		Guidelines: Y_N, Z_N	
Pitch Diameter - Pinion: $D_p =$ 1.800 in		Gear - Number of load cycles: $N_g =$ 4.6E+08		>10' <10'	
Pitch Diameter - Gear: $D_g =$ 5.600 in		Bending Stress Cycle Factor: $Y_{NP} =$ 0.93		1.00 0.93 Fig. 9-22	
Center Distance: $C =$ 3.700 in		Bending Stress Cycle Factor: $Y_{NG} =$ 0.95		1.00 0.95 Fig. 9-22	
Pitch Line Speed: $v_t =$ 565 ft/min		Pitting Stress Cycle Factor: $Z_{NP} =$ 0.89		1.00 0.89 Fig. 9-23	
Transmitted Load: $W_t =$ 292 lb		Pitting Stress Cycle Factor: $Z_{NG} =$ 0.92		1.00 0.92 Fig. 9-23	
Secondary Input Data:		Stress Analysis: Bending			
Face Width Guidelines (in): Min 0.800 Nom 1.200 Max 1.600		Pinion: Required $s_{at} =$ 18,543 psi		See Fig. 9-11 or Table 9-5	
Enter: Face Width: $F =$ 1.250 in		Gear: Required $s_{at} =$ 14,522 psi		Table 9-5	
Ratio: Face width/pinion diameter: $F/D_p =$ 0.69		Stress Analysis: Pitting			
Recommended range of ratio: 0.50 < $F/D_p < 2.00$		Pinion: Required $s_{ac} =$ 143,088 psi		See Fig. 9-12 or Table 9-5	
Enter: Elastic Coefficient: $C_p =$ 2300 Table 9-10		Gear: Required $s_{ac} =$ 138,422 psi		Table 9-5	
Enter: Quality Number: $A_v =$ 11 Table 9-3		Required hardness of pinion HB: 354 Equations in Fig. 9-12-Grade 1			
Dynamic Factor: $K_v =$ 1.32 Table 9-9		Required hardness of gear HB: 340 Equations in Fig. 9-12-Grade 1			
[Factors for computing K_v :] $B =$ 0.826 $C =$ 59.75		Specify materials, alloy and heat treatment, for most severe requirement.			
Reference: $N_p =$ 18 $N_g =$ 56		One possible material specification:			
Bending Geometry Factor-Pinion: $J_p =$ 0.320 Fig. 9-15		Pinion requires HB 354; SAE 4140 OQT 900; HB 388, 16% Elongation			
Bending Geometry Factor-Gear: $J_g =$ 0.400 Fig. 9-15		Gear requires HB 340; SAE 4140 OQT 1000; HB 340, 18% Elongation			
Reference: $m_g =$ 3.11		Comments:			
Enter: Pitting Geometry Factor: $I =$ 0.100 Fig. 9-21		It would be reasonable to specify the same heat treatment for both the pinion and the gear because their contact stresses are very similar.			

DESIGN OF SPUR GEARS

APPLICATION: Problem 61		Factors In Design Analysis:	
Milling machine driven by an electric motor		Alignment Factor, $K_m = 1.0 + C_{pf} + C_{ma}$	If $F < 1.0$ 0.025 0.038 [0.50 < $F/D_p < 2.00$] If $F > 1.0$
Initial Input Data:		Pinion Proportion Factor, C_{pf}	Enter: $C_{pf} = 0.038$ Figure 9-16
Overload Factor: $K_o = 1.50$ Table 9-7	Transmitted Power: $P = 20$ hp	Type of gearing: Open Commer. Ex. Prec.	
Design Power $P_{des} = 30$ hp	Diametral Pitch: $P_d = 6$ Fig. 9-24	Mesh Alignment Factor, $C_{ma} = 0.280$	0.158 0.093 0.058
Input Speed: $n_P = 550$ rpm	Number of Pinion Teeth: $N_P = 24$	Enter: $C_{ma} = 0.093$ Figure 9-17	
Desired Output Speed: $n_G = 185$ rpm	Computed number of gear teeth: $N_G = 71.4$	Alignment Factor: $K_m = 1.13$ [Computed]	
Enter: Chosen No. of Gear Teeth: $N_G = 71$	Computed data:	Size Factor: $K_s = 1.00$ Table 9-8: Use 1.00 if $P_d \geq 5$	
Actual Output Speed: $n_G = 185.9$ rpm	Gear Ratio: $m_G = 2.96$	Pinion Rim Thickness Factor: $K_{sp} = 1.00$ Fig. 9-18: Use 1.00 if solid blank	
Pitch Diameter - Pinion: $D_P = 4.000$ in	Pitch Diameter - Gear: $D_G = 11.833$ in	Gear Rim Thickness Factor: $K_{sg} = 1.00$ Fig. 9-18: Use 1.00 if solid blank	
Center Distance: $C = 7.917$ in	Pitch Line Speed: $v_t = 576$ ft/min	Service Factor: $SF = 1.00$ Use 1.00 if no unusual conditions	
Transmitted Load: $W_t = 1146$ lb	Secondary Input Data:	Reliability Factor: $K_R = 1.25$ Table 9-11 Use 1.00 for $R = .99$	
Face Width Guidelines (in): 1.333 2.000 2.867	Min Norm Max	Enter: Design Life: 20000 hours See Table 9-12	
Enter: Face Width: $F = 2.000$ in	Ratio: Face width/pinion diameter: $F/D_P = 0.50$	Pinion - Number of load cycles: $N_P = 6.6E+08$	Guidelines: Y_N, Z_N
Recommended range of ratio: $0.50 < F/D_P < 2.00$	Enter: Elastic Coefficient: $C_p = 2300$ Table 9-10	Gear - Number of load cycles: $N_G = 2.2E+08$	10' cycles >10' <10'
Enter: Quality Number: $A_v = 9$ Table 9-3	Dynamic Factor: $K_v = 1.20$ Table 9-9	Bending Stress Cycle Factor: $Y_{NP} = 0.94$ Fig. 9-22	1.00 0.94 Fig. 9-22
[Factors for computing K_v]: $B = 0.630$ $C = 70.71$	Reference: $N_P = 24$ $N_G = 71$	Bending Stress Cycle Factor: $Y_{NG} = 0.96$ Fig. 9-22	1.00 0.96 Fig. 9-22
Bending Geometry Factor-Pinion: $J_P = 0.350$ Fig. 9-15	Bending Geometry Factor-Gear: $J_G = 0.420$ Fig. 9-15	Pitting Stress Cycle Factor: $Z_{NP} = 0.91$ Fig. 9-23	1.00 0.91 Fig. 9-23
Reference: $m_G = 2.96$	Enter: Pitting Geometry Factor: $I = 0.108$ Fig. 9-21	Pitting Stress Cycle Factor: $Z_{NG} = 0.93$ Fig. 9-23	1.00 0.93 Fig. 9-23
Stress Analysis: Bending		Pinion: Required $s_{at} = 26,640$ psi See Fig. 9-11 or	
Stress Analysis: Pitting		Gear: Required $s_{at} = 21,737$ psi Table 9-5	
Required hardness of pinion HB: 420 Equations in Fig. 9-12-Grade 1		Pinion: Required $s_{ac} = 164,319$ psi See Fig. 9-12 or	
Required hardness of gear HB: 409 Equations in Fig. 9-12-Grade 1		Gear: Required $s_{ac} = 160,785$ psi Table 9-5	
Specify materials, alloy and heat treatment, for most severe requirement.			
One possible material specification:			
Pinion and gear require flame or induction hardening			
SAE 4140 OQT 800; HB 352, 21% Elongation-Core: Case harden to HRC 50 min.			
Comments:			
It would be reasonable to specify the same heat treatment for both the pinion and the gear because their contact stresses are very similar.			

DESIGN OF SPUR GEARS

APPLICATION: Problem 62		Factors in Design Analysis:	
Punch press driven by an electric motor		Alignment Factor, $K_m = 1.0 + C_{pf} + C_{ma}$	If $F < 1.0$ 0.025 If $F > 1.0$ 0.050 [0.50 < F/D_p < 2.00]
Initial Input Data:		Pinion Proportion Factor, C_{pf}	Enter: $C_{pf} = 0.05$ Figure 9-16
Overload Factor: $K_o = 1.75$ Table 9-7	Transmitted Power: $P = 50$ hp	Type of gearing: Open	Commer. Precision Ex. Prec. 0.173 0.105 0.068
Design Power $P_{des} = 87.5$ hp	Diametral Pitch: $P_d = 4$ Fig. 9-24	Mesh Alignment Factor, $C_{ma} = 0.286$	Enter: $C_{ma} = 0.173$ Figure 9-17
Input Speed: $n_P = 900$ rpm	Number of Pinion Teeth: $N_P = 24$	Alignment Factor: $K_m = 1.22$ [Computed]	
Desired Output Speed: $n_G = 227.5$ rpm	Computed number of gear teeth: 94.9	Size Factor: $K_s = 1.05$ Table 9-8: Use 1.00 if $P_d \geq 5$	
Enter: Chosen No. of Gear Teeth: $N_G = 95$		Pinion Rim Thickness Factor: $K_{RP} = 1.00$ Fig. 9-18: Use 1.00 if solid blank	
		Gear Rim Thickness Factor: $K_{RG} = 1.00$ Fig. 9-18: Use 1.00 if solid blank	
		Service Factor: $SF = 1.00$ Use 1.00 if no unusual conditions	
		Reliability Factor: $K_R = 1.25$ Table 9-11 Use 1.00 for $R = .99$	
Computed data:		Enter: Design Life: 20000 hours	See Table 9-12
Actual Output Speed: $n_G = 227.4$ rpm	Gear Ratio: $m_G = 3.96$	Pinion - Number of load cycles: $N_P = 1.1E+09$	Guidelines: Y_N, Z_N
Pitch Diameter - Pinion: $D_P = 6.000$ in	Pitch Diameter - Gear: $D_G = 23.750$ in	Gear - Number of load cycles: $N_G = 2.7E+08$	10 ⁷ cycles >10 ⁷ <10 ⁷
Center Distance: $C = 14.875$ in	Pitch Line Speed: $v_t = 1414$ ft/min	Bending Stress Cycle Factor: $Y_{NP} = 0.94$	1.00 0.94 Fig. 9-22
Transmitted Load: $W_t = 1167$ lb		Bending Stress Cycle Factor: $Y_{NG} = 0.96$	1.00 0.96 Fig. 9-22
		Pitting Stress Cycle Factor: $Z_{NP} = 0.90$	1.00 0.90 Fig. 9-23
		Pitting Stress Cycle Factor: $Z_{NG} = 0.93$	1.00 0.93 Fig. 9-23
Secondary Input Data:		Stress Analysis: Bending	
Face Width Guidelines (in): 2.000 3.000 4.000	Enter: Face Width: $F = 3.000$ in	Pinion: Required $s_{at} = 19,333$ psi	See Fig. 9-11 or Table 9-5
Ratio: Face width/pinion diameter: $F/D_P = 0.50$	Recommended range of ratio: $0.50 < F/D_P < 2.00$	Gear: Required $s_{at} = 16,226$ psi	
		Stress Analysis: Pitting	
		Pinion: Required $s_{ac} = 139,718$ psi	See Fig. 9-12 or Table 9-5
		Gear: Required $s_{ac} = 135,211$ psi	
		Required hardness of pinion HB: 344 Equations in Fig. 9-12-Grade 1	
		Required hardness of gear HB: 330 Equations in Fig. 9-12-Grade 1	
		Specify materials, alloy and heat treatment, for most severe requirement.	
		One possible material specification:	
		Pinion requires HB 344; SAE 1040 WQT 800; HB 352; 21% elongation	
		Gear requires HB 330; SAE 1040 WQT 800; HB 352; 21% elongation	
		Comments:	
		It would be reasonable to specify the same heat treatment for both the pinion and the gear because their contact stresses are very similar.	

DESIGN OF SPUR GEARS			
APPLICATION: Problem 63		Factors In Design Analysis:	
Cement mixer driven by a gasoline engine			
Initial Input Data:			
Overload Factor: $K_o =$	2.00	Table 9-7	
Transmitted Power: $P =$	2.5 hp		
Design Power $P_{des} =$	5 hp		
Diametral Pitch: $P_d =$	8	Fig. 9-24	
Input Speed: $n_P =$	900 rpm		
Number of Pinion Teeth: $N_P =$	18		
Desired Output Speed: $n_G =$	75 rpm		
Computed number of gear teeth:	216.0		
Enter: Chosen No. of Gear Teeth: $N_G =$	216		
Computed data:			
Actual Output Speed: $n_G =$	75.0 rpm		
Gear Ratio: $m_G =$	12.00		
Pitch Diameter - Pinion: $D_P =$	2.250 in		
Pitch Diameter - Gear: $D_G =$	27.000 in		
Center Distance: $C =$	14.625 in		
Pitch Line Speed: $v_t =$	530 ft/min		
Transmitted Load: $W_t =$	156 lb		
Secondary Input Data:			
Face Width Guidelines (in):	Min 1.000	Nom 1.500	Max 2.000
Enter: Face Width: $F =$	2.250	in	
Ratio: Face width/pinion diameter: $F/D_P =$	1.00		
Recommended range of ratio: $0.50 < F/D_P < 2.00$			
Enter: Elastic Coefficient: $C_p =$	2100	Table 9-10	
Enter: Quality Number: $A_v =$	12	Table 9-3	
Dynamic Factor: $K_v =$	1.38	Table 9-9	
[Factors for computing K_v]: $B =$	0.915	$C = 54.74$	
Reference: $N_P =$	18	$N_G = 216$	
Bending Geometry Factor-Pinion: $J_P =$	0.325	Fig. 9-15	
Bending Geometry Factor-Gear: $J_G =$	0.430	Fig. 9-15	
Reference: $m_G =$	12.00		
Enter: Pitting Geometry Factor: $I =$	0.116	Fig. 9-21	
Alignment Factor, $K_m = 1.0 + C_{pf} + C_{ma}$			
Pinion Proportion Factor, $C_{pf} =$		0.075	0.091
Enter: $C_{pf} =$		0.091	Figure 9-16
Type of gearing:	Open	Commer.	Precision
Mesh Alignment Factor, $C_{ma} =$	0.284	0.162	0.096
Enter: $C_{ma} =$	0.284	Figure 9-17	
Alignment Factor: $K_m =$	1.38	[Computed]	
Size Factor: $K_s =$	1.00	Table 9-8: Use 1.00 if $P_d > 5$	
Pinion Rim Thickness Factor: $K_{sp} =$	1.00	Fig. 9-18: Use 1.00 if solid blank	
Gear Rim Thickness Factor: $K_{sg} =$	1.00	Fig. 9-18: Use 1.00 if solid blank	
Service Factor: $SF =$	1.00	Use 1.00 if no unusual conditions	
Reliability Factor: $K_R =$	1.00	Table 9-11 Use 1.00 for $R = .99$	
Enter: Design Life: 8000 hours		See Table 9-12	
Pinion - Number of load cycles: $N_P =$	4.3E+08	Guidelines: Y_N, Z_N	
Gear - Number of load cycles: $N_G =$	3.6E+07	10 ⁷ cycles >10 ⁷ <10 ⁸	
Bending Stress Cycle Factor: $Y_{NP} =$	0.95	1.00	0.95
Bending Stress Cycle Factor: $Y_{NG} =$	0.99	1.00	0.99
Pitting Stress Cycle Factor: $Z_{NP} =$	0.92	1.00	0.92
Pitting Stress Cycle Factor: $Z_{NG} =$	0.97	1.00	0.97
Stress Analysis: Bending			
Pinion: Required $s_{at} =$	6,796 psi	See Fig. 9-11 or	
Gear: Required $s_{at} =$	4,929 psi	Table 9-5	
Stress Analysis: Pitting			
Pinion: Required $s_{ac} =$	72,362 psi	See Fig. 9-12 or	
Gear: Required $s_{ac} =$	68,632 psi	Table 9-5	
Required hardness of pinion HB:	134	Equations in Fig. 9-12-Grade 1	
Required hardness of gear HB:	123	Equations in Fig. 9-12-Grade 1	
Specify materials, alloy and heat treatment, for most severe requirement.			
One possible material specification:			
Pinion requires less than HB 180; SAE 1040 CD; HB 160; 12% elongation			
Gear requires grey cast iron, ASTM A48, Class 40 [Table 9-6]			
Comments:			
Large gear can be conveniently cast and affixed to the drum of the cement mixer.			
Steel gear can be mounted on engine shaft.			

DESIGN OF SPUR GEARS

APPLICATION: Problem 64		Factors In Design Analysis:			
Wood chipper driven by a gasoline engine: Speed Increase		Alignment Factor, $K_m = 1.0 + C_{pf} + C_{ma}$ If $F < 1.0$ If $F > 1.0$			
Initial Input Data:		Pinion Proportion Factor, $C_{pf} = 0.065$ 0.090 $[0.50 < F/D_p < 2.00]$			
Overload Factor: $K_o = 2.75$ Table 9-7		Enter: $C_{pf} = 0.09$ Figure 9-16			
Transmitted Power: $P = 75$ hp		Type of gearing: Open Commer. Precision Ex. Prec.			
Design Power $P_{des} = 206.25$ hp		Mesh Alignment Factor, $C_{ma} = 0.296$ 0.173 0.105 0.068			
Diametral Pitch: $P_d = 6$ Fig. 9-24		Enter: $C_{ma} = 0.296$ Figure 9-17			
Input Speed: $n_p = 2200$ rpm		Alignment Factor: $K_m = 1.39$ [Computed]			
Number of Pinion Teeth: $N_p = 41$		Size Factor: $K_s = 1.00$ Table 9-8: Use 1.00 if $P_d \geq 5$			
Desired Output Speed: $n_g = 4550$ rpm		Pinion Rim Thickness Factor: $K_{sp} = 1.00$ Fig. 9-18: Use 1.00 if solid blank			
Computed number of gear teeth: 19.8		Gear Rim Thickness Factor: $K_{sg} = 1.00$ Fig. 9-18: Use 1.00 if solid blank			
Enter: Chosen No. of Gear Teeth: $N_g = 20$		Service Factor: $SF = 1.00$ Use 1.00 if no unusual conditions			
Computed data: Note - Gear drives pinion		Reliability Factor: $K_R = 1.00$ Table 9-11 Use 1.00 for $R = .99$			
Actual Output Speed: $n_g = 4510.0$ rpm		Enter: Design Life: 8000 hours See Table 9-12			
Gear Ratio: $m_g = 2.05$		Pinion - Number of load cycles: $N_p = 1.1E+09$ Guidelines: Y_N, Z_N			
Pitch Diameter - Pinion: $D_p = 6.833$ in		Gear - Number of load cycles: $N_g = 5.2E+08$ $> 10^7$ $< 10^7$			
Pitch Diameter - Gear: $D_g = 3.333$ in		Bending Stress Cycle Factor: $Y_{NP} = 0.94$ 1.00 0.94 Fig. 9-22			
Center Distance: $C = 5.083$ in		Bending Stress Cycle Factor: $Y_{NG} = 0.95$ 1.00 0.95 Fig. 9-22			
Pitch Line Speed: $v_t = 3936$ ft/min		Pitting Stress Cycle Factor: $Z_{NP} = 0.90$ 1.00 0.90 Fig. 9-23			
Transmitted Load: $W_t = 629$ lb		Pitting Stress Cycle Factor: $Z_{NG} = 0.91$ 1.00 0.91 Fig. 9-23			
Secondary Input Data:		Stress Analysis: Bending - Note - Gear drives pinion			
Face Width Guidelines (in): Min 1.333 Nom 2.000 Max 2.667		Gear: Required $s_{at} = 24,281$ psi See Fig. 9-11 or Table 9-5			
Enter: Face Width: $F = 3.000$ in		Pinion: Required $s_{at} = 27,666$ psi Table 9-5			
Ratio: Face width/pinion diameter: $F/D_p = 0.90$		Stress Analysis: Pitting - Adjusted equation to use D_g in place of D_p			
Recommended range of ratio: $0.50 < F/D_p < 2.00$		Pinion: Required $s_{ac} = 171,762$ psi See Fig. 9-12 or Table 9-5			
Enter: Elastic Coefficient: $C_p = 2300$ Table 9-10		Gear: Required $s_{ac} = 169,875$ psi Table 9-5			
Enter: Quality Number: $A_v = 11$ Table 9-3		Required hardness of pinion HB: 443 Equations in Fig. 9-12-Grade 1			
Dynamic Factor: $K_v = 1.81$ Table 9-9		Required hardness of gear HB: 437 Equations in Fig. 9-12-Grade 1			
[Factors for computing K_v]: $B = 0.826$ $C = 59.75$		Specify materials, alloy and heat treatment, for most severe requirement.			
Reference: $N_p = 41$ $N_g = 20$		One possible material specification:			
Bending Geometry Factor-Pinion: $J_p = 0.380$ Fig. 9-15		Pinion and gear require flame or induction hardening			
Bending Geometry Factor-Gear: $J_g = 0.330$ Fig. 9-15		SAE 4140 OQT 800; HB 352, 21% Elongation-Core: Case harden to HRC 54 min.			
Reference: $m_g = 2.05$		Comments:			
Enter: Pitting Geometry Factor: $I = 0.096$ Fig. 9-21		It would be reasonable to specify the same heat treatment for both the pinion and the gear because their contact stresses are very similar.			

DESIGN OF SPUR GEARS		APPLICATION:		NOTE: SI Metric data																																	
Initial Input Data: Input Power: $P = 3.0$ kW Input Speed: $n_P = 600$ rpm [See Table 8-3] Module: $m = 3.00$ mm Number of Pinion Teeth: $N_P = 20$ Desired Output Speed: $n_G = 175$ rpm Computed number of gear teeth: 68.6 Enter: Chosen No. of Gear Teeth: $N_G = 68$		Small tractor driven by a gasoline engine Problem 65																																			
Computed data: Actual Output Speed: $n_G = 176.5$ rpm Gear Ratio: $m_G = 3.40$ Pitch Diameter - Pinion: $D_P = 60.00$ mm Pitch Diameter - Gear: $D_G = 204.00$ mm Center Distance: $C = 132.00$ mm Pitch Line Speed: $v_t = 1.88$ m/s Transmitted Load: $W_t = 1592$ N		Factors In Design Analysis: Alignment Factor, $K_m = 1.0 + C_{pf} + C_{ma}$ Pinion Proportion Factor, $C_{pf} = 0.042$ if $F < 25$ [0.049 if $F > 25$ mm] Enter: $C_{pf} = 0.049$ Figure 9-16 Type of gearing: Open Commer. Precision Ex. Prec. Mesh Alignment Factor, $C_{ma} = 0.274$ 0.152 0.088 0.054 Enter: $C_{ma} = 0.274$ Figure 9-17 Alignment Factor: $K_m = 1.32$ [Computed]																																			
Secondary Input Data: <table border="1"> <thead> <tr> <th></th> <th>Min</th> <th>Nom</th> <th>Max</th> </tr> </thead> <tbody> <tr> <td>mm</td> <td>24</td> <td>36</td> <td>48</td> </tr> <tr> <td>Enter: Face Width: $F = 40.0$ mm</td> <td></td> <td></td> <td></td> </tr> <tr> <td>Ratio: Face width/pinion diameter: $F/D_P = 0.67$</td> <td></td> <td></td> <td></td> </tr> <tr> <td>Recommended range of ratio: $0.50 < F/D_P < 2.00$</td> <td></td> <td></td> <td></td> </tr> <tr> <td>Enter: Elastic Coefficient: $C_p = 191$</td> <td></td> <td></td> <td></td> </tr> <tr> <td>Enter: Quality Number: $A_v = 12$</td> <td></td> <td></td> <td></td> </tr> <tr> <td>REF: $N_P, N_G = 20$</td> <td></td> <td></td> <td></td> </tr> </tbody> </table>			Min	Nom	Max	mm	24	36	48	Enter: Face Width: $F = 40.0$ mm				Ratio: Face width/pinion diameter: $F/D_P = 0.67$				Recommended range of ratio: $0.50 < F/D_P < 2.00$				Enter: Elastic Coefficient: $C_p = 191$				Enter: Quality Number: $A_v = 12$				REF: $N_P, N_G = 20$				Overload Factor: $K_o = 2.00$ Table 9-7 Size Factor: $K_s = 1.00$ Table 9-8: Use 1.00 if $P_d \geq 5$ Pinion Rim Thickness Factor: $K_{RP} = 1.00$ Fig. 9-18: Use 1.00 if solid blank Gear Rim Thickness Factor: $K_{RG} = 1.00$ Fig. 9-18: Use 1.00 if solid blank Dynamic Factor: $K_v = 1.32$ [Computed: See Fig. 9-19] Service Factor: $SF = 1.00$ Use 1.00 if no unusual conditions Reliability Factor: $K_R = 1.00$ Table 9-11 Use 1.00 for $R = .99$ Enter: Design Life: 5000 hours See Table 9-12 Pinion - Number of load cycles: $N_P = 1.8E+08$ Guidelines: Y_N, Z_N Gear - Number of load cycles: $N_G = 5.3E+07$ 10 ⁷ cycles > 10 ⁷ < 10 ⁷ Bending Stress Cycle Factor: $Y_{NP} = 0.97$ 1.00 0.97 Fig. 9-22 Bending Stress Cycle Factor: $Y_{NG} = 0.99$ 1.00 0.99 Fig. 9-22 Pitting Stress Cycle Factor: $Z_{NP} = 0.94$ 1.00 0.94 Fig. 9-23 Pitting Stress Cycle Factor: $Z_{NG} = 0.96$ 1.00 0.96 Fig. 9-23			
	Min	Nom	Max																																		
mm	24	36	48																																		
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Enter: Quality Number: $A_v = 12$																																					
REF: $N_P, N_G = 20$																																					
Stress Analysis: Bending Pinion: Required $s_{at} = 144$ MPa See Fig. 9-11 or Gear: Required $s_{at} = 113$ MPa Table 9-5 Stress Analysis: Pitting Pinion: Required $s_{ac} = 958$ MPa See Fig. 9-12 or Gear: Required $s_{ac} = 938$ MPa Table 9-5 Specify materials, alloy and heat treatment, for most severe requirement. One possible material specification: Steel pinion, Steel gear Pinion requires HB 341; SAE 4340 OQT 1000; HB 363 Gear requires HB 332; SAE 4340 OQT 1000; HB 363 (Same as pinion)		For K_v : B 0.915 C 3.90 Through-Hardened Grade 1 Steel HB 105 Fig. 9-11 HB 46 Fig. 9-11 HB 341 Fig. 9-12 HB 332 Fig. 9-12																																			

DESIGN OF SPUR GEARS		NOTE: SI Metric data		APPLICATION:	
Initial Input Data: Input Power: $P = 75.0$ kW Input Speed: $n_P = 4500$ rpm [See Table 8-3] Module: $m = 4.00$ mm Number of Pinion Teeth: $N_P = 24$ Desired Output Speed: $n_G = 3600$ rpm Computed number of gear teeth: 30.0 Enter: Chosen No. of Gear Teeth: $N_G = 30$				Electric power generator driven by a water turbine Problem 66	
Computed data: Actual Output Speed: $n_G = 3600.0$ rpm Gear Ratio: $m_G = 1.25$ Pitch Diameter - Pinion: $D_P = 96.00$ mm Pitch Diameter - Gear: $D_G = 120.00$ mm Center Distance: $C = 108.00$ mm Pitch Line Speed: $v_t = 22.62$ m/s Transmitted Load: $W_t = 3316$ N				Factors In Design Analysis: Alignment Factor, $K_m = 1.0 + C_{pf} + C_{ma}$ Pinion Proportion Factor, $C_{pf} = 0.027$ if $F < 25$ if $F > 25$ mm Enter: $C_{pf} = 0.040$ Figure 9-16 Type of gearing: Open Commer. Precision Ex. Prec. Mesh Alignment Factor, $C_{ma} = 0.280$ 0.158 0.093 0.058 Enter: $C_{ma} = 0.158$ Figure 9-17 Alignment Factor: $K_m = 1.20$ [Computed]	
Overload Factor: $K_o = 1.20$ Table 9-7 Size Factor: $K_s = 1.00$ Table 9-8: Use 1.00 if $P_d \geq 5$ Pinion Rim Thickness Factor: $K_{RP} = 1.00$ Fig. 9-18: Use 1.00 if solid blank Gear Rim Thickness Factor: $K_{RG} = 1.00$ Fig. 9-18: Use 1.00 if solid blank				For K_v : B 0.397 C 5.97	
Dynamic Factor: $K_v = 1.26$ [Computed: See Fig. 9-19] Service Factor: $S_F = 1.00$ Use 1.00 if no unusual conditions Reliability Factor: $K_R = 1.00$ Table 9-11 Use 1.00 for $R = .99$				Enter: Design Life: 100000 hours See Table 9-12 Pinion - Number of load cycles: $N_P = 2.7E+10$ Guidelines: Y_N, Z_N Gear - Number of load cycles: $N_G = 2.2E+10$ 10 ⁷ cycles >10 ⁷ <10 ⁷	
Bending Stress Cycle Factor: $Y_{NP} = 0.88$ Fig. 9-22 Bending Stress Cycle Factor: $Y_{NG} = 0.89$ Fig. 9-22 Pitting Stress Cycle Factor: $Z_{NP} = 0.83$ Fig. 9-23 Pitting Stress Cycle Factor: $Z_{NG} = 0.84$ Fig. 9-23				Through-Hardened Grade 1 Steel HB 19 Fig. 9-11 HB 8 Fig. 9-11 HB 310 Fig. 9-12 HB 305 Fig. 9-12	
Stress Analysis: Bending Pinion: Required $s_{at} = 98$ MPa See Fig. 9-11 or Gear: Required $s_{at} = 93$ MPa Table 9-5					
Stress Analysis: Pitting Pinion: Required $s_{ac} = 889$ MPa See Fig. 9-12 or Gear: Required $s_{ac} = 878$ MPa Table 9-5					
Specify materials, alloy and heat treatment, for most severe requirement. One possible material specification: Steel pinion, Steel gear Pinion requires HB 310; SAE 4340 OQT 1100; HB 321 Gear requires HB 305; SAE 4340 OQT 1100; HB 321 (Same as pinion)					

DESIGN OF SPUR GEARS	
APPLICATION: Problem 67 Commercial band saw driven by an electric motor	
Initial Input Data: Overload Factor: $K_o = 1.50$ Table 9-7 Transmitted Power: $P = 12$ hp Design Power $P_{des} = 18$ hp Diametral Pitch: $P_d = 10$ Fig. 9-24 Input Speed: $n_P = 3450$ rpm Number of Pinion Teeth: $N_P = 18$ Desired Output Speed: $n_G = 730$ rpm Computed number of gear teeth: 85.1 Enter: Chosen No. of Gear Teeth: $N_G = 85$	
Computed data: Actual Output Speed: $n_G = 730.6$ rpm Gear Ratio: $m_G = 4.72$ Pitch Diameter - Pinion: $D_P = 1.800$ in Pitch Diameter - Gear: $D_G = 8.500$ in Center Distance: $C = 5.150$ in Pitch Line Speed: $v_t = 1626$ ft/min Transmitted Load: $W_t = 244$ lb	
Secondary Input Data: Face Width Guidelines (in): Min 0.800 Norm 1.200 Max 1.600 Enter: Face Width: $F = 1.250$ in Ratio: Face width/pinion diameter: $F/D_P = 0.69$ Recommended range of ratio: $0.50 < F/D_P < 2.00$ Enter: Elastic Coefficient: $C_p = 2300$ Table 9-10 Enter: Quality Number: $A_v = 9$ Table 9-3 Dynamic Factor: $K_v = 1.33$ Table 9-9 [Factors for computing K_v]: $B = 0.630$ $C = 70.71$ Reference: $N_P = 18$ $N_G = 85$ Bending Geometry Factor-Pinion: $J_P = 0.310$ Fig. 9-15 Bending Geometry Factor-Gear: $J_G = 0.400$ Fig. 9-15 Reference: $m_G = 4.72$ Enter: Pitting Geometry Factor: $I = 0.106$ Fig. 9-21	
Factors in Design Analysis: Alignment Factor, $K_m = 1.0 + C_{pf} + C_{ma}$ If $F < 1.0$ 0.044 If $F > 1.0$ 0.048 [0.50 < $F/D_P < 2.00$] Pinion Proportion Factor, $C_{pf} = 0.048$ Figure 9-16 Enter: $C_{pf} = 0.048$ Type of gearing: Open Commer. Precision Ex. Prec. Mesh Alignment Factor, $C_{ma} = 0.268$ 0.147 0.083 0.051 Enter: $C_{ma} = 0.147$ Figure 9-17 Alignment Factor: $K_m = 1.20$ [Computed]	
Size Factor: $K_s = 1.00$ Table 9-8: Use 1.00 if $P_d \geq 5$ Pinion Rim Thickness Factor: $K_{RP} = 1.00$ Fig. 9-18: Use 1.00 if solid blank Gear Rim Thickness Factor: $K_{RG} = 1.00$ Fig. 9-18: Use 1.00 if solid blank Service Factor: $SF = 1.00$ Use 1.00 if no unusual conditions Reliability Factor: $K_R = 1.00$ Table 9-11 Use 1.00 for $R = .99$	
Enter: Design Life: 8000 hours See Table 9-12 Pinion - Number of load cycles: $N_P = 1.7E+09$ Guidelines: Y_N, Z_N Gear - Number of load cycles: $N_G = 3.5E+08$ 10' cycles >10' <10'	
Bending Stress Cycle Factor: $Y_{NP} = 0.93$ 1.00 0.93 Fig. 9-22 Bending Stress Cycle Factor: $Y_{NG} = 0.96$ 1.00 0.96 Fig. 9-22 Pitting Stress Cycle Factor: $Z_{NP} = 0.89$ 1.00 0.89 Fig. 9-23 Pitting Stress Cycle Factor: $Z_{NG} = 0.92$ 1.00 0.92 Fig. 9-23	
Stress Analysis: Bending Pinion: Required $s_{at} = 16,101$ psi See Fig. 9-11 or Gear: Required $s_{at} = 12,088$ psi Table 9-5	
Stress Analysis: Pitting Pinion: Required $s_{ac} = 127,467$ psi See Fig. 9-12 or Gear: Required $s_{ac} = 123,310$ psi Table 9-5	
Required hardness of pinion HB: 305 Equations in Fig. 9-12-Grade 1 Required hardness of gear HB: 293 Equations in Fig. 9-12-Grade 1 Specify materials, alloy and heat treatment, for most severe requirement.	
One possible material specification: Pinion requires HB 305; SAE 4140 OQT 1100; HB 321, 19% Elongation Gear requires HB 293; SAE 4140 OQT 1200; HB 283, 20% Elongation	
Comments: It would be reasonable to specify the same heat treatment for both the pinion and the gear because their contact stresses are very similar.	

DESIGN OF SPUR GEARS

APPLICATION: [Problem 68]		Factors in Design Analysis:			
Commercial band saw driven by an electric motor		Alignment Factor, $K_m = 1.0 + C_{pf} + C_{ma}$	If $F < 1.0$	If $F > 1.0$	
Initial Input Data:		Pinion Proportion Factor, $C_{pf} =$	0.063	0.064	[0.50 < $F/D_p < 2.00$]
Overload Factor: $K_o =$	1.50 Table 9-7	Enter: $C_{ma} =$	0.064	Figure 9-16	
Transmitted Power: $P =$	12 hp	Type of gearing:	Open	Commer.	Ex. Prec.
Design Power $P_{des} =$	18 hp	Mesh Alignment Factor, $C_{ma} =$	0.266	0.145	0.082 0.049
Diametral Pitch: $P_d =$	14 Fig. 9-24	Enter: $C_{ms} =$	0.145	Figure 9-17	
Input Speed: $n_P =$	3450 rpm	Alignment Factor: $K_m =$	1.21	[Computed]	
Number of Pinion Teeth: $N_P =$	18	Size Factor: $K_s =$	1.00	Table 9-8: Use 1.00 if $P_d \geq 5$	
Desired Output Speed: $n_G =$	730 rpm	Pinion Rim Thickness Factor: $K_{ap} =$	1.00	Fig. 9-18: Use 1.00 if solid blank	
Computed number of gear teeth: 85.1		Gear Rim Thickness Factor: $K_{ag} =$	1.00	Fig. 9-18: Use 1.00 if solid blank	
Enter: Chosen No. of Gear Teeth: $N_G =$	85	Service Factor: $SF =$	1.00	Use 1.00 if no unusual conditions	
Computed data:		Reliability Factor: $K_R =$	1.00	Table 9-11 Use 1.00 for $R = .99$	
Actual Output Speed: $n_G =$	730.6 rpm	Enter: Design Life: 8000 hours	See Table 9-12		
Gear Ratio: $m_G =$	4.72	Pinion - Number of load cycles: $N_P = 1.7E+09$	Guidelines: Y_N, Z_N		
Pitch Diameter - Pinion: $D_P =$	1.286 in	Gear - Number of load cycles: $N_G = 3.5E+08$	10 ⁷ cycles	>10 ⁷	<10 ⁷
Pitch Diameter - Gear: $D_G =$	6.071 in	Bending Stress Cycle Factor: $Y_{NP} =$	0.93	1.00	0.93 Fig. 9-22
Center Distance: $C =$	3.679 in	Bending Stress Cycle Factor: $Y_{NG} =$	0.96	1.00	0.96 Fig. 9-22
Pitch Line Speed: $v_t =$	1161 ft/min	Pitting Stress Cycle Factor: $Z_{NP} =$	0.89	1.00	0.89 Fig. 9-23
Transmitted Load: $W_t =$	341 lb	Pitting Stress Cycle Factor: $Z_{NG} =$	0.92	1.00	0.92 Fig. 9-23
Secondary Input Data:		Stress Analysis: Bending			
Face Width Guidelines (in): 0.571 0.857 1.143	Min Nom Max	Pinion: Required $s_{at} =$	30,567 psi	See Fig. 9-11 or	
Enter: Face Width: $F =$	1.125 in	Gear: Required $s_{at} =$	22,949 psi	Table 9-5	
Ratio: Face width/pinion diameter: $F/D_P =$	0.88	Stress Analysis: Pitting			
Recommended range of ratio: 0.50 < $F/D_P < 2.00$		Pinion: Required $s_{ac} =$	175,629 psi	See Fig. 9-12 or	
Enter: Elastic Coefficient: $C_p =$	2300 Table 9-10	Gear: Required $s_{ac} =$	169,902 psi	Table 9-5	
Enter: Quality Number: $A_v =$	7 Table 9-3	Required hardness of pinion HB: 455 Equations in Fig. 9-12-Grade 1			
Dynamic Factor: $K_v =$	1.15 Table 9-9	Required hardness of gear HB: 437 Equations in Fig. 9-12-Grade 1			
[Factors for computing K_v]: $B =$	0.397 $C =$	Specify materials, alloy and heat treatment, for most severe requirement.			
Reference: $N_P =$	18 $N_G =$	One possible material specification:			
Bending Geometry Factor-Pinion: $J_P =$	0.310 Fig. 9-15	Pinion requires case hardening by carburizing			
Bending Geometry Factor-Gear: $J_G =$	0.400 Fig. 9-15	Gear requires case hardening by carburizing			
Reference: $m_G =$	4.72	Specifications: Example selection			
Enter: Pitting Geometry Factor: $I =$	0.106 Fig. 9-21	Specify SAE 4620 DOQT 300; Case hardness HRC 62; ductile core; HB 248			
For both pinion and gear					

DESIGN OF SPUR GEARS

APPLICATION: Problem 69		Factors in Design Analysis:	
Machine tool driven by an electric motor		Alignment Factor, $K_m = 1.0 + C_{pf} + C_{ma}$	If $F < 1.0$ If $F > 1.0$
Initial Input Data:		Pinion Proportion Factor, $C_{pf} = 0.048$	0.060 [0.50 < $F/D_p < 2.00$]
Overload Factor: $K_o = 1.50$	Table 9-7	Enter: $C_{ma} = 0.060$	Figure 9-16
Transmitted Power: $P = 20$ hp		Type of gearing: Open	Commer. Precision Ex. Prec.
Design Power $P_{des} = 30$ hp		Mesh Alignment Factor, $C_{ma} = 0.280$	0.158 0.093 0.058
Diametral Pitch: $P_d = 8$	Fig. 9-24	Enter: $C_{ms} = 0.093$	Figure 9-17
Input Speed: $n_P = 650$ rpm		Alignment Factor: $K_m = 1.15$	[Computed]
Number of Pinion Teeth: $N_P = 22$		Size Factor: $K_s = 1.00$	Table 9-8: Use 1.00 if $P_d \geq 5$
Desired Output Speed: $n_G = 112.5$ rpm		Pinion Rim Thickness Factor: $K_{RP} = 1.00$	Fig. 9-18: Use 1.00 if solid blank
Computed number of gear teeth: 127.1		Gear Rim Thickness Factor: $K_{RG} = 1.00$	Fig. 9-18: Use 1.00 if solid blank
Enter: Chosen No. of Gear Teeth: $N_G = 128$		Service Factor: $SF = 1.00$	Use 1.00 if no unusual conditions
Computed data:		Reliability Factor: $K_R = 1.00$	Table 9-11 Use 1.00 for $R = .99$
Actual Output Speed: $n_G = 111.7$ rpm		Enter: Design Life: 25000 hours	See Table 9-12
Gear Ratio: $m_G = 5.82$		Pinion - Number of load cycles: $N_P = 9.8E+08$	Guidelines: Y_N, Z_N
Pitch Diameter - Pinion: $D_P = 2.750$ in		Gear - Number of load cycles: $N_G = 1.7E+08$	10 ⁷ cycles >10 ⁷ <10 ⁷
Pitch Diameter - Gear: $D_G = 16.000$ in		Bending Stress Cycle Factor: $Y_{NP} = 0.94$	1.00 0.94 Fig. 9-22
Center Distance: $C = 9.375$ in		Bending Stress Cycle Factor: $Y_{NG} = 0.97$	1.00 0.97 Fig. 9-22
Pitch Line Speed: $v_t = 468$ ft/min		Pitting Stress Cycle Factor: $Z_{NP} = 0.90$	1.00 0.90 Fig. 9-23
Transmitted Load: $W_t = 1410$ lb		Pitting Stress Cycle Factor: $Z_{NG} = 0.94$	1.00 0.94 Fig. 9-23
Secondary Input Data:		Stress Analysis: Bending	
Face Width Guidelines (in): Min 1.000 Nom 1.500 Max 2.000		Pinion: Required $s_{at} = 32,958$ psi	See Fig. 9-11 or Table 9-5
Enter: Face Width: $F = 2.000$ in		Gear: Required $s_{at} = 25,043$ psi	
Ratio: Face width/pinion diameter: $F/D_P = 0.73$		Stress Analysis: Pitting	
Recommended range of ratio: $0.50 < F/D_P < 2.00$		Pinion: Required $s_{ac} = 173,012$ psi	See Fig. 9-12 or Table 9-5
Enter: Elastic Coefficient: $C_p = 2300$	Table 9-10	Gear: Required $s_{ac} = 165,650$ psi	
Enter: Quality Number: $A_v = 7$	Table 9-3	Required hardness of pinion HB: 447 Equations in Fig. 9-12-Grade 1	
Dynamic Factor: $K_v = 1.10$	Table 9-9	Required hardness of gear HB: 424 Equations in Fig. 9-12-Grade 1	
[Factors for computing K_v]: $B = 0.397$ $C = 83.77$		Specify materials, alloy and heat treatment, for most severe requirement.	
Reference: $N_P = 22$ $N_G = 128$		One possible material specification:	
Bending Geometry Factor-Pinion: $J_P = 0.345$	Fig. 9-15	Pinion requires case hardening by carburizing	
Bending Geometry Factor-Gear: $J_G = 0.440$	Fig. 9-15	Gear requires case hardening by carburizing	
Reference: $m_G = 5.82$		Specifications: Example selection	
Enter: Pitting Geometry Factor: $I = 0.106$	Fig. 9-21	Specify SAE 4620 DOQT 300; Case hardness HRC 62; ductile core; HB 248	
		For both pinion and gear	

DESIGN OF SPUR GEARS

APPLICATION: Problem 70		Factors in Design Analysis:	
Crane cable drum driven by an electric motor		Alignment Factor, $K_m = 1.0 + C_{pf} + C_{ma}$	If $F < 1.0$ 0.067 0.087 If $F > 1.0$ [0.50 < $F/D_p < 2.00$]
Initial Input Data:		Pinion Proportion Factor, $C_{pf} =$ 0.067 0.087 Figure 9-16	
Overload Factor: $K_o =$ 1.50 Table 9-7		Enter: $C_{ma} =$ 0.087 Figure 9-16	
Transmitted Power: $P =$ 25 hp		Type of gearing: Open Commer. Precision Ex. Prec.	
Design Power $P_{des} =$ 37.5 hp		Mesh Alignment Factor, $C_{ma} =$ 0.290 0.167 0.100 0.064	
Diametral Pitch: $P_d =$ 6 Fig. 9-24		Enter: $C_{ma} =$ 0.167 Figure 9-17	
Input Speed: $n_P =$ 925 rpm		Alignment Factor: $K_m =$ 1.25 [Computed]	
Number of Pinion Teeth: $N_P =$ 17		Size Factor: $K_s =$ 1.00 Table 9-8: Use 1.00 if $P_d \geq 5$	
Desired Output Speed: $n_G =$ 163 rpm		Pinion Rim Thickness Factor: $K_{BP} =$ 1.00 Fig. 9-18: Use 1.00 if solid blank	
Computed number of gear teeth: 96.5		Gear Rim Thickness Factor: $K_{BG} =$ 1.00 Fig. 9-18: Use 1.00 if solid blank	
Enter: Chosen No. of Gear Teeth: $N_G =$ 96		Service Factor: $SF =$ 1.00 Use 1.00 if no unusual conditions	
Computed data:		Reliability Factor: $K_R =$ 1.00 Table 9-11 Use 1.00 for $R = .99$	
Actual Output Speed: $n_G =$ 163.8 rpm		Enter: Design Life: 31200 hours See Table 9-12	
Gear Ratio: $m_G =$ 5.65		Pinion - Number of load cycles: $N_P = 1.7E+09$ Guidelines: Y_N, Z_N	
Pitch Diameter - Pinion: $D_P =$ 2.833 in		Gear - Number of load cycles: $N_G = 3.1E+08$ 10 ⁷ cycles > 10 ⁷ < 10 ⁸	
Pitch Diameter - Gear: $D_G =$ 16.000 in		Bending Stress Cycle Factor: $Y_{NP} =$ 0.93 1.00 0.93 Fig. 9-22	
Center Distance: $C =$ 9.417 in		Bending Stress Cycle Factor: $Y_{NG} =$ 0.96 1.00 0.96 Fig. 9-22	
Pitch Line Speed: $v_t =$ 686 ft/min		Pitting Stress Cycle Factor: $Z_{NP} =$ 0.89 1.00 0.89 Fig. 9-23	
Transmitted Load: $W_t =$ 1202 lb		Pitting Stress Cycle Factor: $Z_{NG} =$ 0.92 1.00 0.92 Fig. 9-23	
Secondary Input Data:		Stress Analysis: Bending	
Face Width Guidelines (in): 1.333 2.000 2.667	Min Nom Max	Pinion: Required $s_{at} =$ 21,194 psi See Fig. 9-11 or	
Enter: Face Width: $F =$ 2.600 in		Gear: Required $s_{at} =$ 14,421 psi Table 9-5	
Ratio: Face width/pinion diameter: $F/D_P =$ 0.92		Stress Analysis: Pitting	
Recommended range of ratio: 0.50 < $F/D_P < 2.00$		Pinion: Required $s_{ac} =$ 144,763 psi See Fig. 9-12 or	
Enter: Elastic Coefficient: $C_p =$ 2300 Table 9-10		Gear: Required $s_{ac} =$ 140,043 psi Table 9-5	
Enter: Quality Number: $A_v =$ 7 Table 9-3		Required hardness of pinion HB: 359 Equations in Fig. 9-12-Grade 1	
Dynamic Factor: $K_v =$ 1.11 Table 9-9		Required hardness of gear HB: 345 Equations in Fig. 9-12-Grade 1	
[Factors for computing K_v]: $B =$ 0.397 $C =$ 83.77		Specify materials, alloy and heat treatment, for most severe requirement.	
Reference: $N_P =$ 17 $N_G =$ 96		One possible material specification:	
Bending Geometry Factor-Pinion: $J_P =$ 0.295 Fig. 9-15		Pinion requires HB 359; SAE 8650 OQT 1000; HB 363; 14% elongation	
Bending Geometry Factor-Gear: $J_G =$ 0.420 Fig. 9-15		Pinion requires HB 345; SAE 8650 OQT 1000; HB 363; 14% elongation	
Reference: $m_G =$ 5.65		Comments:	
Enter: Pitting Geometry Factor: $I =$ 0.109 Fig. 9-21			

SPUR GEARS		APPLICATION: Centrifugal pump driven by an electric motor Chapter 9-Problem 71	
POWER TRANSMITTING CAPACITY			
Initial Input Data:			
Enter: Face Width:	$F = 1.250$ in		
Input Speed:	$n_p = 1725$ rpm		
Diametral Pitch:	$P_d = 10$		
Number of Pinion Teeth:	$N_p = 25$		
Number of Gear Teeth:	$N_g = 60$		
Computed data:			
Actual Output Speed:	$n_g = 718.8$ rpm		
Gear Ratio:	$m_g = 2.40$		
Pitch Diameter - Pinion:	$D_p = 2.500$ in		
Pitch Diameter - Gear:	$D_g = 6.000$ in		
Center Distance:	$C = 4.250$ in		
Pitch Line Speed:	$v_t = 1129$ ft/min		
Transmitted Load at P_{min} Capacity:	$W_t = 377$ lb		
Power Transmitting Capacity: (Using Eq. 9-32, 9-34)			
Pinion: Based on Bending Stress:	25.08 hp		
Gear: Based on Bending Stress:	27.27 hp		
Pinion: Based on Contact Stress:	14.21 hp		
Gear: Based on Contact Stress:	12.91 hp		
Power Transmitting Capacity: 12.91 hp			
Enter: Elastic Coefficient:	$C_p = 2300$	Table 9-10	
Enter: Quality Number:	$A_v = 9$	Table 9-3	
REF: $N_p, N_g = 25, 60$			
Enter: Bending Geometry Factors: Press. angle = 20 deg			
Pinion:	$J_p = 0.363$	Fig. 9-15	
Gear:	$J_g = 0.415$	Fig. 9-15	
Enter: Pitting Geometry Factor:			
	$I = 0.104$	Fig. 9-21	
REF: $m_g = 2.40$			
Factors in Design Analysis:			
Alignment Factor, $K_m = 1.0 + C_{pt} + C_{ma}$	If $F < 1.0$	If $F > 1.0$	$F/D_p = 0.50$
Pinion Proportion Factor, C_{pt}	0.025	0.028	$0.50 < F/D_p < 2.00$
Enter: $C_{pt} = 0.028$	Figure 9-16		
Type of gearing:	Open	Commer.	Precision
Mesh Alignment Factor, C_{ma}	0.268	0.147	0.083
Enter: $C_{ma} = 0.147$	Figure 9-17		
Alignment Factor: $K_m = 1.18$	[Computed]		
Overload Factor: $K_o = 1.50$	Table 9-7		
Size Factor: $K_s = 1.00$	Table 9-8: Use 1.00 if $P_d \geq 5$		
Pinion Rim Thickness Factor: $K_{ap} = 1.00$	Fig. 9-18: Use 1.00 if solid blank		
Gear Rim Thickness Factor: $K_{ag} = 1.00$	Fig. 9-18: Use 1.00 if solid blank		
Dynamic Factor: $K_v = 1.28$	[Computed: See Fig. 9-21]		
Service Factor: $SF = 1.00$	Use 1.00 if no unusual conditions		
Reliability Factor: $K_R = 1.00$	Table 9-8 Use 1.00 for $R = .99$		
Enter: Design Life: 15000 hours See Table 9-7			
Pinion - Number of load cycles: $N_p = 1.6E+09$	Guidelines: Y_N, Z_N		
Gear - Number of load cycles: $N_g = 6.5E+08$	10 ⁷ cycles $> 10^7$ $< 10^7$		
Bending Stress Cycle Factor: $Y_{NP} = 0.93$	1.00	0.93	Fig. 9-22
Bending Stress Cycle Factor: $Y_{NG} = 0.94$	1.00	0.94	Fig. 9-22
Pitting Stress Cycle Factor: $Z_{NP} = 0.89$	1.00	0.89	Fig. 9-24
Pitting Stress Cycle Factor: $Z_{NG} = 0.91$	1.00	0.91	Fig. 9-24
Allowable Bending Stress Numbers: (Input)			
Pinion: $S_{at} = 39,100$ psi	See Fig. 9-11 or		
Gear: $S_{at} = 36,800$ psi	Table 9-5		
Allowable Contact Stress Numbers: (Input)			
Pinion: $S_{ac} = 138,600$ psi	See Fig. 9-12 or		
Gear: $S_{ac} = 129,200$ psi	Table 9-5		
Material specification: Steel pinion; Steel gear: through hardened			
Pinion material: SAE 4140 OQT 1000	340 HB		
Gear material: SAE 4140 OQT 1100	311 HB		

SPUR GEARS		APPLICATION: Heavy duty conveyor driven by a gasoline engine	
POWER TRANSMITTING CAPACITY		Chapter 9-Problem 72	
Initial Input Data: Enter: Face Width: $F = 2.000$ in Input Speed: $n_P = 1500$ rpm Diametral Pitch: $P_d = 6$ Number of Pinion Teeth: $N_P = 35$ Number of Gear Teeth: $N_G = 100$			
Computed data: Actual Output Speed: $n_G = 525.0$ rpm Gear Ratio: $m_G = 2.86$ Pitch Diameter - Pinion: $D_P = 5.833$ in Pitch Diameter - Gear: $D_G = 16.667$ in Center Distance: $C = 11.250$ in Pitch Line Speed: $v_t = 2291$ ft/min Transmitted Load at P_{min} Capacity: $W_t = 277$ lb			
Power Transmitting Capacity: (Using Eq. 9-32, 9-34) Pinion: Based on Bending Stress: 90.79 hp Gear: Based on Bending Stress: 21.63 hp Pinion: Based on Contact Stress: 86.50 hp Gear: Based on Contact Stress: 19.26 hp Power Transmitting Capacity: 19.26 hp			
Enter: Elastic Coefficient: $C_p = 2100$ Table 9-10 Enter: Quality Number: $A_v = 11$ Table 9-3 REF: $N_P, N_G = 35, 100$			
Enter: Bending Geometry Factors: Press. angle = 20 deg Pinion: $J_P = 0.410$ Fig. 9-15 Gear: $J_G = 0.450$ Fig. 9-15 Enter: Pitting Geometry Factor: $I = 0.114$ Fig. 9-21 REF: $m_G = 2.86$			
Factors in Design Analysis: Alignment Factor, $K_m = 1.0 + C_{pt} + C_{ma}$ If $F < 1.0$ 0.025 If $F > 1.0$ 0.038 [0.50 < $F/D_P < 2.00$] Pinion Proportion Factor, $C_{pt} = 0.038$ Figure 9-16 Enter: $C_{pt} = 0.038$ Figure 9-16 Type of gearing: Open Commer. Precision Ex. Prec. Mesh Alignment Factor, $C_{ma} = 0.280$ 0.158 0.093 0.058 Enter: $C_{ma} = 0.158$ Figure 9-17 Alignment Factor: $K_m = 1.20$ [Computed]			
Overload Factor: $K_o = 2.00$ Table 9-7 Size Factor: $K_s = 1.00$ Table 9-8: Use 1.00 if $P_d \geq 5$ Pinion Rim Thickness Factor: $K_{gp} = 1.00$ Fig. 9-18: Use 1.00 if solid blank Gear Rim Thickness Factor: $K_{gg} = 1.00$ Fig. 9-18: Use 1.00 if solid blank Dynamic Factor: $K_v = 1.63$ [Computed: See Fig. 9-21]			
Service Factor: $SF = 1.00$ Use 1.00 if no unusual conditions Reliability Factor: $K_R = 1.00$ Table 9-8 Use 1.00 for $R = 99$			
Enter: Design Life: 15000 hours See Table 9-7 Pinion - Number of load cycles: $N_P = 1.4E+09$ Guidelines: Y_N, Z_N Gear - Number of load cycles: $N_G = 4.7E+08$ 10 ⁷ cycles > 10 ⁷ < 10 ⁷			
Bending Stress Cycle Factor: $Y_{NP} = 0.93$ 1.00 0.93 Fig. 9-22 Bending Stress Cycle Factor: $Y_{NG} = 0.95$ 1.00 0.95 Fig. 9-22 Pitting Stress Cycle Factor: $Z_{NP} = 0.89$ 1.00 0.89 Fig. 9-24 Pitting Stress Cycle Factor: $Z_{NG} = 0.92$ 1.00 0.92 Fig. 9-24			
Allowable Bending Stress Numbers: (Input) Pinion: $S_{at} = 40,000$ psi See Fig. 9-11 Gear: $S_{at} = 8,500$ psi Table 9-6			
Allowable Contact Stress Numbers: (Input) Pinion: $S_{ac} = 142,400$ psi See Fig. 9-12 Gear: $S_{ac} = 65,000$ psi Table 9-6			
Material specification: Steel pinion; Steel gear: through hardened Pinion material: SAE 1040 WQT 800 352 HB Gear: Gray cast iron, ASTM A48, Class 30			

For K_v :

B	0.826
A	59.75

Through-Hardened	Grade 1 Steel
40.0 ksi	Fig. 9-11
42.8 ksi	Fig. 9-11
142.4 ksi	Fig. 9-12
29.1 ksi	Fig. 9-12

SPUR GEARS		APPLICATION:	
POWER TRANSMITTING CAPACITY		Heavy duty conveyor driven by a gasoline engine Chapter 9-Problem 73 - Redesign of system in Problem 72 to get capacity > 25 hp	
Initial Input Data:		Factors in Design Analysis:	
Enter: Face Width: $F = 2.420$ in		Alignment Factor: $K_m = 1.0 + C_{pf} + C_{ma}$	if $F < 1.0$ if $F > 1.0$ $F/D_p = 0.50$
Input Speed: $n_p = 1500$ rpm		Pinion Proportion Factor: $C_{pf} = 0.025$	0.043 [0.50 < F/D_p < 2.00]
Diametral Pitch: $P_d = 6$		Enter: $C_{pf} = 0.043$	Figure 9-16
Number of Pinion Teeth: $N_p = 35$		Type of gearing: Open	Commer. Precision Ex. Prec.
Number of Gear Teeth: $N_g = 100$		Mesh Alignment Factor: $C_{ma} = 0.287$	0.165 0.098 0.062
		Enter: $C_{ma} = 0.165$	Figure 9-17
		Alignment Factor: $K_m = 1.21$	[Computed]
Computed data:		Overload Factor: $K_o = 2.00$	Table 9-7
Actual Output Speed: $n_g = 525.0$ rpm		Size Factor: $K_s = 1.00$	Table 9-8: Use 1.00 if $P_d \geq 5$
Gear Ratio: $m_g = 2.86$		Pinion Rim Thickness Factor: $K_{ap} = 1.00$	Fig. 9-18: Use 1.00 if solid blank
Pitch Diameter - Pinion: $D_p = 5.833$ in		Gear Rim Thickness Factor: $K_{ag} = 1.00$	Fig. 9-18: Use 1.00 if solid blank
Pitch Diameter - Gear: $D_g = 16.667$ in		Dynamic Factor: $K_v = 1.50$	[Computed: See Fig. 9-21]
Center Distance: $C = 11.250$ in		Service Factor: $SF = 1.00$	Use 1.00 if no unusual conditions
Pitch Line Speed: $v_t = 2291$ ft/min		Reliability Factor: $K_R = 1.00$	Table 9-8 Use 1.00 for $R = 99$
Transmitted Load at P_{min} Capacity: $W_t = 361$ lb		Enter: Design Life: 15000 hours	See Table 9-7
Power Transmitting Capacity: (Using Eq. 9-32, 9-34)		Pinion - Number of load cycles: $N_p = 1.4E+09$	Guidelines: Y_N, Z_N
Pinion: Based on Bending Stress: 118.13 hp		Gear - Number of load cycles: $N_g = 4.7E+08$	10 ⁶ cycles > 10 ⁷ < 10 ⁸
Gear: Based on Bending Stress: 28.14 hp		Bending Stress Cycle Factor: $Y_{NP} = 0.93$	1.00 0.93 Fig. 9-22
Pinion: Based on Contact Stress: 112.55 hp		Bending Stress Cycle Factor: $Y_{NG} = 0.95$	1.00 0.95 Fig. 9-22
Gear: Based on Contact Stress: 25.06 hp		Pitting Stress Cycle Factor: $Z_{NP} = 0.89$	1.00 0.89 Fig. 9-24
Power Transmitting Capacity: 25.06 hp		Pitting Stress Cycle Factor: $Z_{NG} = 0.92$	1.00 0.92 Fig. 9-24
Enter: Elastic Coefficient: $C_p = 2100$	Table 9-10	Allowable Bending Stress Numbers: (Input)	
Enter: Quality Number: $A_v = 10$	Table 9-3	Pinion: $s_{at} = 40,000$ psi	See Fig. 9-11
REF: $N_p, N_g = 35, 100$		Gear: $s_{at} = 8,500$ psi	Table 9-6
Enter: Bending Geometry Factors: Press. angle = 20 deg		Allowable Contact Stress Numbers: (Input)	
Pinion: $J_p = 0.410$	Fig. 9-15	Pinion: $s_{ac} = 142,400$ psi	See Fig. 9-12
Gear: $J_g = 0.450$	Fig. 9-15	Gear: $s_{ac} = 65,000$ psi	Table 9-6
Enter: Pitting Geometry Factor: $I = 0.114$	Fig. 9-21	Material specification: Steel pinion; Steel gear: through hardened	
REF: $m_g = 2.86$		Pinion material: SAE 1040 WQT 800 352 HB	
Note: Increased face width from 2.00 to 2.42 in.		Gear: Gray cast iron, ASTM A48, Class 30	
Changed quality number from $A_v = 11$ to 10 (More precise)			

For K_v :	
B	0.731
A	65.04

Through-Hardened	
Grade 1 Steel	
40.0 ksi	Fig. 9-11
42.8 ksi	Fig. 9-11
142.4 ksi	Fig. 9-12
29.1 ksi	Fig. 9-12

DESIGN OF SPUR GEARS			
APPLICATION: Problem 74 - First pair Assembly conveyor driven by an electric motor			
Double reduction - First pair - Input Data:			
Overload Factor:	$K_o = 1.50$	Table 9-7	
Transmitted Power:	$P = 10$ hp		
Design Power P_{des} :	15 hp		
Diametral Pitch:	$P_d = 8$	Fig. 9-24	
Input Speed:	$n_P = 1750$ rpm		
Number of Pinion Teeth:	$N_P = 18$		
Desired Output Speed:	$n_G = 425$ rpm		
Computed number of gear teeth:	74.1		
Enter: Chosen No. of Gear Teeth:	$N_G = 75$		
Computed data:			
Actual Output Speed:	$n_G = 420.0$ rpm		
Gear Ratio:	$m_G = 4.17$		
Pitch Diameter - Pinion:	$D_P = 2.250$ in		
Pitch Diameter - Gear:	$D_G = 9.375$ in		
Center Distance:	$C = 5.813$ in		
Pitch Line Speed:	$v_t = 1031$ ft/min		
Transmitted Load:	$W_t = 320$ lb		
Secondary Input Data:			
Face Width Guidelines (in):	Min 1.000 1.500 2.000	Max	
Enter: Face Width:	$F = 1.500$ in		
Ratio: Face width/pinion diameter:	$F/D_P = 0.67$		
Recommended range of ratio:	$0.50 < F/D_P < 2.00$		
Enter: Elastic Coefficient:	$C_p = 2300$	Table 9-10	
Enter: Quality Number:	$A_v = 11$	Table 9-3	
Dynamic Factor:	$K_v = 1.43$	Table 9-9	
[Factors for computing K_v]:	$B = 0.826$ $C = 59.75$		
Reference:	$N_P = 18$ $N_G = 75$		
Bending Geometry Factor-Pinion:	$J_P = 0.315$	Fig. 9-15	
Bending Geometry Factor-Gear:	$J_G = 0.410$	Fig. 9-15	
Reference:	$m_G = 4.17$		
Enter: Pitting Geometry Factor:	$I = 0.106$	Fig. 9-21	
Note: Larger part of the total reduction (4.17) in this pair Higher diametral pitch - 8 compared to 6 in pair 2			
Factors in Design Analysis:			
Alignment Factor, $K_m = 1.0 + C_{pf} + C_{ma}$	If $F < 1.0$ 0.042 0.048 0.050 $< F/D_P < 2.00$ If $F > 1.0$		
Pinion Proportion Factor, C_{pf}	Enter: $C_{pf} = 0.048$ Figure 9-16 Type of gearing: Open Commer. Precision Ex. Prec. Mesh Alignment Factor, $C_{ma} = 0.272$ 0.150 0.086 0.053 Enter: $C_{ma} = 0.150$ Figure 9-17 Alignment Factor: $K_m = 1.20$ [Computed]		
Size Factor:	$K_s = 1.00$	Table 9-8: Use 1.00 if $P_d \geq 5$	
Pinion Rim Thickness Factor:	$K_{BP} = 1.00$	Fig. 9-18: Use 1.00 if solid blank	
Gear Rim Thickness Factor:	$K_{BG} = 1.00$	Fig. 9-18: Use 1.00 if solid blank	
Service Factor:	$SF = 1.00$	Use 1.00 if no unusual conditions	
Reliability Factor:	$K_R = 1.00$	Table 9-11 Use 1.00 for $R = .99$	
Enter: Design Life: 15000 hours See Table 9-12			
Pinion - Number of load cycles:	$N_P = 1.6E+09$	Guidelines: Y_N, Z_N	
Gear - Number of load cycles:	$N_G = 3.8E+08$	10 ⁷ cycles $> 10^7$ $< 10^7$	
Bending Stress Cycle Factor:	$Y_{NP} = 0.93$	1.00 0.93	Fig. 9-22
Bending Stress Cycle Factor:	$Y_{NG} = 0.95$	1.00 0.95	Fig. 9-22
Pitting Stress Cycle Factor:	$Z_{NP} = 0.89$	1.00 0.89	Fig. 9-23
Pitting Stress Cycle Factor:	$Z_{NG} = 0.92$	1.00 0.92	Fig. 9-23
Stress Analysis: Bending			
Pinion: Required s_{at}	$14,940$ psi	See Fig. 9-11 or	
Gear: Required s_{at}	$11,237$ psi	Table 9-5	
Stress Analysis: Pitting			
Pinion: Required s_{ac}	$123,772$ psi	See Fig. 9-12 or	
Gear: Required s_{ac}	$119,736$ psi	Table 9-5	
Required hardness of pinion HB:	294	Equations in Fig. 9-12-Grade 1	
Required hardness of gear HB:	281	Equations in Fig. 9-12-Grade 1	
Specify materials, alloy and heat treatment, for most severe requirement.			
One possible material specification:			
Pinion requires HB 294; SAE 4340 OQT 1100; HB 321; 19% elongation Gear requires HB 281; SAE 4340 OQT 1200; HB 293; 20% elongation			
Comments:			

DESIGN OF SPUR GEARS			
APPLICATION: Problem 74 - Second pair			
Assembly conveyor driven by an electric motor			
Double reduction - Second pair - Input Data:			
Overload Factor:	$K_o = 1.50$	Table 9-7	
Transmitted Power:	$P = 10$ hp		
Design Power P_{des} :	15 hp		
Diametral Pitch:	$P_d = 6$	Fig. 9-24	
Input Speed:	$n_P = 420$ rpm		
Number of Pinion Teeth:	$N_P = 18$		
Desired Output Speed:	$n_G = 148$ rpm		
Computed number of gear teeth:	51.1		
Enter: Chosen No. of Gear Teeth:	$N_G = 51$		
Computed data:			
Actual Output Speed:	$n_G = 148.2$ rpm		
Gear Ratio:	$m_G = 2.83$		
Pitch Diameter - Pinion:	$D_P = 3.000$ in		
Pitch Diameter - Gear:	$D_G = 8.500$ in		
Center Distance:	$C = 5.750$ in		
Pitch Line Speed:	$v_t = 330$ ft/min		
Transmitted Load:	$W_t = 1000$ lb		
Secondary Input Data:			
Face Width Guidelines (in):	Min 1.333 2.000 Max 2.667		
Enter: Face Width:	$F = 2.000$ in		
Ratio: Face width/pinion diameter:	$F/D_P = 0.67$		
Recommended range of ratio:	$0.50 < F/D_P < 2.00$		
Enter: Elastic Coefficient:	$C_p = 2300$	Table 9-10	
Enter: Quality Number:	$A_v = 11$	Table 9-3	
Dynamic Factor:	$K_v = 1.25$	Table 9-9	
[Factors for computing K_v]	$B = 0.826$ $C = 59.75$		
Reference:	$N_P = 18$ $N_G = 51$		
Bending Geometry Factor-Pinion:	$J_P = 0.315$	Fig. 9-15	
Bending Geometry Factor-Gear:	$J_G = 0.395$	Fig. 9-15	
Reference:	$m_G = 2.83$		
Enter: Pitting Geometry Factor:	$I = 0.108$	Fig. 9-21	
Note: Smaller part of the total reduction (42.83) in this pair Lower diametral pitch - 6 compared to 8 in pair 1			
Factors in Design Analysis:			
Alignment Factor, $K_m = 1.0 + C_{pf} + C_{ma}$	If $F < 1.0$ 0.042 0.054 [0.50 < $F/D_P < 2.00$] If $F > 1.0$		
Pinion Proportion Factor, C_{pf}	Enter: $C_{pf} = 0.054$ Figure 9-16		
Type of gearing:	Open	Commer.	Precision
Mesh Alignment Factor, C_{ma}	0.280	0.158	0.093
Enter: C_{ma}	0.158	Figure 9-17	
Alignment Factor: K_m	1.21	[Computed]	
Size Factor: K_s	1.00	Table 9-8: Use 1.00 if $P_d \geq 5$	
Pinion Rim Thickness Factor: K_{BP}	1.00	Fig. 9-18: Use 1.00 if solid blank	
Gear Rim Thickness Factor: K_{BG}	1.00	Fig. 9-18: Use 1.00 if solid blank	
Service Factor: SF	1.00	Use 1.00 if no unusual conditions	
Reliability Factor: K_R	1.00	Table 9-11 Use 1.00 for $R = .99$	
Enter: Design Life:	15000 hours	See Table 9-12	
Pinion - Number of load cycles: N_P	$3.8E+08$	Guidelines: Y_N, Z_N	
Gear - Number of load cycles: N_G	$1.3E+08$	10^7 cycles $> 10^7$ $< 10^7$	
Bending Stress Cycle Factor: Y_{NP}	0.95	1.00	0.95 Fig. 9-22
Bending Stress Cycle Factor: Y_{NG}	0.97	1.00	0.97 Fig. 9-22
Pitting Stress Cycle Factor: Z_{NP}	0.92	1.00	0.92 Fig. 9-23
Pitting Stress Cycle Factor: Z_{NG}	0.94	1.00	0.94 Fig. 9-23
Stress Analysis: Bending			
Pinion: Required s_{at}	$22,702$ psi	See Fig. 9-11 or	
Gear: Required s_{at}	$17,731$ psi	Table 9-5	
Stress Analysis: Pitting			
Pinion: Required s_{ac}	$147,790$ psi	See Fig. 9-12 or	
Gear: Required s_{ac}	$144,645$ psi	Table 9-5	
Required hardness of pinion HB: 369 Equations in Fig. 9-12-Grade 1			
Required hardness of gear HB: 359 Equations in Fig. 9-12-Grade 1			
Specify materials, alloy and heat treatment, for most severe requirement.			
One possible material specification:			
Pinion requires HB 369; SAE 4340 OQT 900; HB 388; 15% elongation			
Gear requires HB 359; SAE 4340 OQT 1000; HB 363; 17% elongation			
Comments:			
Center Distances and sizes of gears are well balanced			

DESIGN OF SPUR GEARS									
APPLICATION: Problem 75 - First pair									
Food waste grinder driven by an electric motor									
Double reduction - First pair - Input Data:									
Overload Factor:	$K_o = 1.50$	Table 9-7							
Transmitted Power:	$P = 0.5$	hp							
Design Power P_{des} :	$P_{des} = 0.75$	hp							
Diametral Pitch:	$P_d = 16$	Fig. 9-24							
Input Speed:	$n_P = 850$	rpm							
Number of Pinion Teeth:	$N_P = 18$								
Desired Output Speed:	$n_G = 190$	rpm							
Computed number of gear teeth:	80.5								
Enter: Chosen No. of Gear Teeth:	$N_G = 81$								
Computed data:									
Actual Output Speed:	$n_G = 188.9$	rpm							
Gear Ratio:	$m_G = 4.50$								
Pitch Diameter - Pinion:	$D_P = 1.125$	in							
Pitch Diameter - Gear:	$D_G = 5.063$	in							
Center Distance:	$C = 3.094$	in							
Pitch Line Speed:	$v_t = 250$	ft/min							
Transmitted Load:	$W_t = 66$	lb							
Secondary Input Data:									
Face Width Guidelines (in):	Min	Nom	Max						
Enter: Face Width:	0.500	0.750	1.000						
Ratio: Face width/pinion diameter:	$F/D_P = 0.44$								
Recommended range of ratio:	$0.50 < F/D_P < 2.00$								
Enter: Elastic Coefficient:	$C_p = 2300$	Table 9-10							
Enter: Quality Number:	$A_v = 9$	Table 9-3							
Dynamic Factor:	$K_v = 1.14$	Table 9-9							
[Factors for computing K_v]:	$B = 0.630$	$C = 70.71$							
Reference:	$N_P = 18$	$N_G = 81$							
Bending Geometry Factor-Pinion:	$J_P = 0.320$	Fig. 9-15							
Bending Geometry Factor-Gear:	$J_G = 0.415$	Fig. 9-15							
Reference:	$m_G = 4.50$								
Enter: Pitting Geometry Factor:	$I = 0.106$	Fig. 9-21							

Factors in Design Analysis:									
Alignment Factor, $K_m = 1.0 + C_{pf} + C_{ma}$	If $F < 1.0$	If $F > 1.0$							
Pinion Proportion Factor, $C_{pf} = 0.019$	0.013	$[0.50 < F/D_P < 2.00]$							
Enter: $C_{pf} = 0.019$	Figure 9-16								
Type of gearing:	Open	Commer.	Precision	Ex. Prec.					
Mesh Alignment Factor, $C_{ma} = 0.255$	0.135	0.074	0.043						
Enter: $C_{ma} = 0.135$	Figure 9-17								
Alignment Factor: $K_m = 1.15$	[Computed]								
Size Factor: $K_s = 1.00$	Table 9-8: Use 1.00 if $P_d \geq 5$								
Pinion Rim Thickness Factor: $K_{BP} = 1.00$	Fig. 9-18: Use 1.00 if solid blank								
Gear Rim Thickness Factor: $K_{BG} = 1.00$	Fig. 9-18: Use 1.00 if solid blank								
Service Factor: $SF = 1.00$	Use 1.00 if no unusual conditions								
Reliability Factor: $K_R = 1.00$	Table 9-11 Use 1.00 for $R = .99$								
Enter: Design Life: 8000 hours	See Table 9-12								
Pinion - Number of load cycles: $N_P = 4.1E+08$	Guidelines: Y_N, Z_N								
Gear - Number of load cycles: $N_G = 9.1E+07$	10^7 cycles	$> 10^7$							
Bending Stress Cycle Factor: $Y_{NP} = 0.95$	1.00	0.95	Fig. 9-22						
Bending Stress Cycle Factor: $Y_{NG} = 0.98$	1.00	0.98	Fig. 9-22						
Pitting Stress Cycle Factor: $Z_{NP} = 0.92$	1.00	0.92	Fig. 9-23						
Pitting Stress Cycle Factor: $Z_{NG} = 0.95$	1.00	0.95	Fig. 9-23						
Stress Analysis: Bending									
Pinion: Required $s_{at} = 13,639$ psi	See Fig. 9-11 or								
Gear: Required $s_{at} = 10,195$ psi	Table 9-5								
Stress Analysis: Pitting									
Pinion: Required $s_{ac} = 116,542$ psi	See Fig. 9-12 or								
Gear: Required $s_{ac} = 112,862$ psi	Table 9-5								
Required hardness of pinion HB:	272	Equations in Fig. 9-12-Grade 1							
Required hardness of gear HB:	260	Equations in Fig. 9-12-Grade 1							
Specify materials, alloy and heat treatment, for most severe requirement.									
One possible material specification:									
Pinion requires HB 272; SAE 4340 OQT 1200; HB 293; 21% elongation									
Gear requires HB 260; SAE 4340 OQT 1200; HB 293; 20% elongation									
Comments:									

Larger face width required for pair 2 (1.15 in) vs. 0.50 in for pair 1
Stresses higher for pair 2 than for pair 1, requiring higher hardness

DESIGN OF SPUR GEARS									
APPLICATION: Problem 75 - Second pair									
Food waste grinder driven by an electric motor									
Double reduction - Second pair - Input Data:									
Overload Factor: $K_o = 1.50$		Table 9-7							
Transmitted Power: $P = 0.5$ hp									
Design Power $P_{des} = 0.75$ hp									
Diametral Pitch: $P_d = 16$		Fig. 9-24							
Input Speed: $n_P = 188.9$ rpm									
Number of Pinion Teeth: $N_P = 18$									
Desired Output Speed: $n_G = 42$ rpm									
Computed number of gear teeth: $N_G = 81.0$									
Enter: Chosen No. of Gear Teeth: $N_G = 81$									
Computed data:									
Actual Output Speed: $n_G = 42.0$ rpm									
Gear Ratio: $m_G = 4.50$									
Pitch Diameter - Pinion: $D_P = 1.125$ in									
Pitch Diameter - Gear: $D_G = 5.063$ in									
Center Distance: $C = 3.094$ in									
Pitch Line Speed: $v_t = 56$ ft/min									
Transmitted Load: $W_t = 297$ lb									
Secondary Input Data:									
Face Width Guidelines (in): Min 0.500, Nom 0.750, Max 1.000									
Enter: Face Width: $F = 1.150$ in									
Ratio: Face width/pinion diameter: $F/D_P = 1.02$									
Recommended range of ratio: $0.50 < F/D_P < 2.00$									
Enter: Elastic Coefficient: $C_p = 2300$		Table 9-10							
Enter: Quality Number: $A_v = 9$		Table 9-3							
Dynamic Factor: $K_v = 1.07$		Table 9-9							
[Factors for computing K_v]: $B = 0.630$		$C = 70.71$							
Reference: $N_P = 18$		$N_G = 81$							
Bending Geometry Factor-Pinion: $J_P = 0.320$		Fig. 9-15							
Bending Geometry Factor-Gear: $J_G = 0.415$		Fig. 9-15							
Reference: $m_G = 4.50$									
Enter: Pitting Geometry Factor: $I = 0.106$		Fig. 9-21							
<p>Note: Equal reduction ratios used for pairs 1 and 2</p> <p>Equal diametral pitches (16) used for both pairs</p>									

FACTORS IN DESIGN ANALYSIS:									
Alignment Factor: $K_m = 1.0 + C_{pf} + C_{ma}$		If $F < 1.0$		If $F > 1.0$					
Pinion Proportion Factor: $C_{pf} = 0.077$		0.077		0.079		[0.50 < $F/D_P < 2.00$]			
Enter: $C_{pf} = 0.079$		Figure 9-16							
Type of gearing: Open		Commer.		Precision		Ex. Prec.			
Mesh Alignment Factor: $C_{ma} = 0.266$		0.145		0.082		0.050			
Enter: $C_{ma} = 0.145$		Figure 9-17							
Alignment Factor: $K_m = 1.22$		[Computed]							
Size Factor: $K_s = 1.00$		Table 9-8: Use 1.00 if $P_d \geq 5$							
Pinion Rim Thickness Factor: $K_{BP} = 1.00$		Fig. 9-18: Use 1.00 if solid blank							
Gear Rim Thickness Factor: $K_{BG} = 1.00$		Fig. 9-18: Use 1.00 if solid blank							
Service Factor: $SF = 1.00$		Use 1.00 if no unusual conditions							
Reliability Factor: $K_R = 1.00$		Table 9-11 Use 1.00 for $R = .99$							
Enter: Design Life: 8000 hours		See Table 9-12							
Pinion - Number of load cycles: $N_P = 9.1E+07$		Guidelines: Y_N, Z_N							
Gear - Number of load cycles: $N_G = 2.0E+07$		10^7 cycles		$> 10^7$		$< 10^7$			
Bending Stress Cycle Factor: $Y_{NP} = 0.98$		1.00		0.98		Fig. 9-22			
Bending Stress Cycle Factor: $Y_{NG} = 1.01$		1.00		1.01		Fig. 9-22			
Pitting Stress Cycle Factor: $Z_{NP} = 0.95$		1.00		0.95		Fig. 9-23			
Pitting Stress Cycle Factor: $Z_{NG} = 0.98$		1.00		0.98		Fig. 9-23			
Stress Analysis: Bending									
Pinion: Required $s_{at} = 25,734$ psi		See Fig. 9-11 or							
Gear: Required $s_{at} = 19,253$ psi		Table 9-5							
Stress Analysis: Pitting									
Pinion: Required $s_{ac} = 157,454$ psi		See Fig. 9-12 or							
Gear: Required $s_{ac} = 152,634$ psi		Table 9-5							
Required hardness of pinion HB: 399		Equations in Fig. 9-12-Grade 1							
Required hardness of gear HB: 384		Equations in Fig. 9-12-Grade 1							
Specify materials, alloy and heat treatment, for most severe requirement.									
One possible material specification:									
Pinion requires HB 399; SAE 4340 OQT 800; HB 415; 12% elongation									
Gear requires HB 384; SAE 4340 OQT 800; HB 415; 12% elongation									
Comments:									
Pinion and gear made from same material and heat treatment for processing convenience.									
Larger face width required for pair 2 (1.15 in) vs. 0.50 in for pair 1									
Stresses higher for pair 2 than for pair 1, requiring higher hardness									

DESIGN OF SPUR GEARS	
APPLICATION: Problem 76 - First pair Small hand drill driven by an electric motor	
Double reduction - First pair - Input Data: Overload Factor: $K_o = 1.50$ Table 9-7 Transmitted Power: $P = 0.25$ hp Design Power $P_{des} = 0.375$ hp Diametral Pitch: $P_d = 24$ Fig. 9-24 Input Speed: $n_P = 3000$ rpm Number of Pinion Teeth: $N_P = 15$ Desired Output Speed: $n_G = 1300$ rpm Computed number of gear teeth: 34.6 Enter: Chosen No. of Gear Teeth: $N_G = 35$	
Computed data: Actual Output Speed: $n_G = 1285.7$ rpm Gear Ratio: $m_G = 2.33$ Pitch Diameter - Pinion: $D_P = 0.625$ in Pitch Diameter - Gear: $D_G = 1.458$ in Center Distance: $C = 1.042$ in Pitch Line Speed: $v_t = 491$ ft/min Transmitted Load: $W_t = 17$ lb	
Secondary Input Data: Face Width Guidelines (in): Min 0.333 Nom 0.500 Max 0.667 Enter: Face Width: $F = 0.250$ in Ratio: Face width/pinion diameter: $F/D_P = 0.40$ Recommended range of ratio: $0.50 < F/D_P < 2.00$	
Enter: Elastic Coefficient: $C_p = 2300$ Table 9-10 Enter: Quality Number: $A_v = 9$ Table 9-3 Dynamic Factor: $K_v = 1.19$ Table 9-9 [Factors for computing K_v]: $B = 0.630$ $C = 70.71$	
Reference: $N_P = 15$ $N_G = 35$ Bending Geometry Factor-Pinion: $J_P = 0.250$ Fig. 9-15 Bending Geometry Factor-Gear: $J_G = 0.355$ Fig. 9-15 Reference: $m_G = 2.33$ Enter: Pitting Geometry Factor: $I = 0.088$ Fig. 9-21	
Note: Equal reduction ratios used for pairs 1 and 2 Equal diametral pitches (24) used for both pairs	
Factors in Design Analysis: Alignment Factor, $K_m = 1.0 + C_{pm} + C_{ma}$ If $F < 1.0$ 0.015 If $F > 1.0$ 0.006 $[0.50 < F/D_P < 2.00]$ Pinion Proportion Factor, $C_{pf} = 0.015$ Figure 9-16 Enter: $C_{pf} = 0.015$ Type of gearing: Open Commer. Precision Ex. Prec. Mesh Alignment Factor, $C_{ma} = 0.251$ 0.131 0.071 0.041 Enter: $C_{ma} = 0.131$ Figure 9-17 Alignment Factor: $K_m = 1.15$ [Computed]	
Size Factor: $K_s = 1.00$ Table 9-8: Use 1.00 if $P_d \geq 5$ Pinion Rim Thickness Factor: $K_{BP} = 1.00$ Fig. 9-18: Use 1.00 if solid blank Gear Rim Thickness Factor: $K_{BG} = 1.00$ Fig. 9-18: Use 1.00 if solid blank Service Factor: $SF = 1.00$ Use 1.00 if no unusual conditions Reliability Factor: $K_R = 1.00$ Table 9-11 Use 1.00 for $R = .99$	
Enter: Design Life: 5000 hours See Table 9-12 Pinion - Number of load cycles: $N_P = 9.0E+08$ Guidelines: Y_N, Z_N Gear - Number of load cycles: $N_G = 3.9E+08$ 10^7 cycles $> 10^7$ $< 10^7$	
Bending Stress Cycle Factor: $Y_{NP} = 0.94$ 1.00 0.94 Fig. 9-22 Bending Stress Cycle Factor: $Y_{NG} = 0.95$ 1.00 0.95 Fig. 9-22 Pitting Stress Cycle Factor: $Z_{NP} = 0.90$ 1.00 0.90 Fig. 9-23 Pitting Stress Cycle Factor: $Z_{NG} = 0.92$ 1.00 0.92 Fig. 9-23	
Stress Analysis: Bending Pinion: Required $s_{at} = 14,014$ psi See Fig. 9-11 or Gear: Required $s_{at} = 9,765$ psi Table 9-5	
Stress Analysis: Pitting Pinion: Required $s_{ac} = 127,649$ psi See Fig. 9-12 or Gear: Required $s_{ac} = 124,874$ psi Table 9-5	
Required hardness of pinion HB: 306 Equations in Fig. 9-12-Grade 1 Required hardness of gear HB: 297 Equations in Fig. 9-12-Grade 1 Specify materials, alloy and heat treatment, for most severe requirement.	
One possible material specification: Pinion requires HB 306; SAE 4340 OQT 1100; HB 321; 19% elongation Gear requires HB 297; SAE 4340 OQT 1100; HB 321; 19% elongation	
Comments:	
Larger face width required for pair 2 (0.56 in) vs. 0.25 in for pair 1 Stresses nearly equal for pair 2 and for pair 1, requiring equal hardnesses	

DESIGN OF SPUR GEARS	
APPLICATION: Problem 76 - Second pair Small hand drill driven by an electric motor	
Double reduction - Second pair - Input Data: Overload Factor: $K_o = 1.50$ Table 9-7 Transmitted Power: $P = 0.25$ hp Design Power $P_{des} = 0.375$ hp Diametral Pitch: $P_d = 24$ Fig. 9-24 Input Speed: $n_P = 1285.7$ rpm Number of Pinion Teeth: $N_P = 15$ Desired Output Speed: $n_G = 550$ rpm Computed number of gear teeth: 35.1 Enter: Chosen No. of Gear Teeth: $N_G = 35$	
Computed data: Actual Output Speed: $n_G = 551.0$ rpm Gear Ratio: $m_G = 2.33$ Pitch Diameter - Pinion: $D_P = 0.625$ in Pitch Diameter - Gear: $D_G = 1.458$ in Center Distance: $C = 1.042$ in Pitch Line Speed: $v_t = 210$ ft/min Transmitted Load: $W_t = 39$ lb	
Secondary Input Data: Face Width Guidelines (in): Min Nom Max Enter: Face Width: $F = 0.560$ in Ratio: Face width/pinion diameter: $F/D_P = 0.90$ Recommended range of ratio: $0.50 < F/D_P < 2.00$ Enter: Elastic Coefficient: $C_p = 2300$ Table 9-10 Enter: Quality Number: $A_v = 9$ Table 9-3 Dynamic Factor: $K_v = 1.12$ Table 9-9 [Factors for computing K_v]: $B = 0.630$ $C = 70.71$ Reference: $N_P = 15$ $N_G = 35$ Bending Geometry Factor-Pinion: $J_P = 0.250$ Fig. 9-15 Bending Geometry Factor-Gear: $J_G = 0.355$ Fig. 9-15 Reference: $m_G = 2.33$ Enter: Pitting Geometry Factor: $I = 0.088$ Fig. 9-21	
Factors in Design Analysis: Alignment Factor, $K_m = 1.0 + C_{pf} + C_{ma}$ If $F < 1.0$ If $F > 1.0$ Pinion Proportion Factor, $C_{pf} = 0.065$ 0.059 $[0.50 < F/D_P < 2.00]$ Enter: $C_{pf} = 0.065$ Figure 9-16 Type of gearing: Open Commer. Precision Ex. Prec. Mesh Alignment Factor, $C_{ma} = 0.256$ 0.136 0.075 0.044 Enter: $C_{ma} = 0.136$ Figure 9-17 Alignment Factor: $K_m = 1.20$ [Computed] Size Factor: $K_s = 1.00$ Table 9-8: Use 1.00 if $P_d \geq 5$ Pinion Rim Thickness Factor: $K_{BP} = 1.00$ Fig. 9-18: Use 1.00 if solid blank Gear Rim Thickness Factor: $K_{BG} = 1.00$ Fig. 9-18: Use 1.00 if solid blank Service Factor: $SF = 1.00$ Use 1.00 if no unusual conditions Reliability Factor: $K_R = 1.00$ Table 9-11 Use 1.00 for $R = .99$ Enter: Design Life: 5000 hours See Table 9-12 Pinion - Number of load cycles: $N_P = 3.9E+08$ Guidelines: Y_N, Z_N Gear - Number of load cycles: $N_G = 1.7E+08$ 10^7 cycles $> 10^7$ $< 10^7$ Bending Stress Cycle Factor: $Y_{NP} = 0.95$ 1.00 0.95 Fig. 9-22 Bending Stress Cycle Factor: $Y_{NG} = 0.97$ 1.00 0.97 Fig. 9-22 Pitting Stress Cycle Factor: $Z_{NP} = 0.92$ 1.00 0.92 Fig. 9-23 Pitting Stress Cycle Factor: $Z_{NG} = 0.94$ 1.00 0.94 Fig. 9-23	
Stress Analysis: Bending Pinion: Required $s_{at} = 14,339$ psi See Fig. 9-11 or Gear: Required $s_{at} = 9,890$ psi Table 9-5	
Stress Analysis: Pitting Pinion: Required $s_{ac} = 126,985$ psi See Fig. 9-12 or Gear: Required $s_{ac} = 124,283$ psi Table 9-5	
Required hardness of pinion HB: 304 Equations in Fig. 9-12-Grade 1 Required hardness of gear HB: 296 Equations in Fig. 9-12-Grade 1 Specify materials, alloy and heat treatment, for most severe requirement. One possible material specification: Pinion requires HB 304; SAE 4340 OQT 1100; HB 321; 19% elongation Gear requires HB 296; SAE 4340 OQT 1100; HB 321; 19% elongation	
Comments:	

Larger face width required for pair 2 (0.56 in) vs. 0.25 in for pair 1
 Stresses nearly equal for pair 2 and for pair 1, requiring equal hardnesses

Note: Equal reduction ratios used for pairs 1 and 2
 Equal diametral pitches (24) used for both pairs

DESIGN OF PLASTIC SPUR GEARS

Application:

Band saw driven by electric motor
Problem 77

Initial Input Data:

Input Power: $P = 0.25$ hp
Input Speed: $n_P = 551$ rpm
Diametral Pitch: $P_d = 12$
Number of Pinion Teeth: $N_P = 18$
Desired Output Speed: $n_G = 159.15$ rpm

Computed number of gear teeth: 62.319

Enter Chosen No. of Gear Teeth: $N_G = 62$

Computed data:

Actual Output Speed: $n_G = 160.0$ rpm
Gear Ratio: $m_G = 3.444$
Pitch Diameter - Pinion: $D_P = 1.500$ in
Pitch Diameter - Gear: $D_G = 5.167$ in
Center Distance: $C = 6.667$ in
Pitch Line Speed: $v_t = 216.4$ ft/min
Transmitted Load: $W_t = 38.11$ lb

Secondary Input Data - Pinion:

Tooth Form: 20 degree full depth
Lewis Form Factor: $Y = 0.521$ Table 9-15
Safety Factor: $SF = 1.50$ Ref. Table 9-7
Material: Unfilled Nylon
Allowable Bending Stress: $s_{at} = 6000$ psi Table 9-14 or Fig. 9-28

Required Face Width: $F = 0.219$ in

Enter Specified Face Width: $F = 0.220$ in

Actual Bending Stress in Pinion: $s_t = 5985$ psi

Secondary Input Data - Gear:

Tooth Form: 20 degree full depth Same as for pinion
Lewis Form Factor: $Y = 0.716$ Table 9-15
Safety Factor: $SF = 1.50$ Same as for pinion
Material: Unfilled Acetal
Allowable Bending Stress: $s_{at} = 5000$ psi Table 9-14 or Fig. 9-28

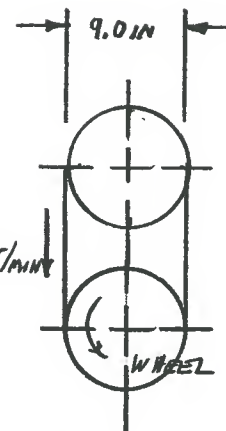
Face Width - Gear: $F = 0.220$ in Same as for pinion

Actual Bending Stress in Gear: $s_t = 4355$ psi Must be $< s_{at}$

From Problem 76

$$N_{\text{WHEEL}} = N_{\text{BLADE}} = 375 \text{ ft/min}$$

To produce blade
speed of 375 ft/min



$$N_{\text{WHEEL}} = \pi D_W n_W / 12$$

$$n_W = \frac{12 N_{\text{WHEEL}}}{\pi D_W} = \frac{12(375)}{\pi(9 \text{ in})}$$

$$n_W = 159.2 \text{ RPM}$$

CONNECT WHEEL ON OUTPUT
SHAFT OF GEAR REDUCER.
ONE PAIR OF GEARS.

78.

RACK DRIVES FURNACE DOOR. $N_{\text{RACK}} \geq 2.0 \text{ FT/S} = 120 \text{ FT/MIN}$

$$N_R = N_{E_D} = \pi D_D m_D / 12$$

$$m_D \geq \frac{12 N_{E_D}}{\pi D_D} = \frac{(12)(120)}{\pi D_D} = \frac{458.4}{D_D}$$

POSSIBLE VALUES FOR D_G :

D_D	m_D	$TV = m_A/m_D$
10.0 IN	45.84 RPM	32.72
9.0 IN	50.93	29.45
12.167	37.76	39.72 USED THIS

PAIR 1: $m_A = 1500 \text{ RPM}$ $m_B = 172.3 \text{ RPM}$

$$N_A = 17, N_B = 148, VR_1 = 8.71$$

PAIR 2: $m_C = m_B = 172.3 \text{ RPM}$ $m_D = 37.76$

$$N_C = 16, N_D = 73, VR_2 = 4.56$$

SEE TWO SPREADSHEET SOLUTIONS

DESIGNED FOR OPEN GEARING

DESIGN LIFE: IN EACH CYCLE, RACK MOVES 6 FT EACH WAY

TOTAL OF 12 FT FOR FULL CYCLE. AT 2.0 FT/S, CYCLE TIME IS:

$$\frac{1 \text{ SEC}}{2.0 \text{ FT}} \times 12 \text{ FT} = 6.0 \text{ SEC/CYCLE}$$

$$\frac{6.0 \text{ SEC}}{\text{CYCLE}} \times \frac{6 \text{ CYCLES}}{\text{Hr}} \times \frac{24 \text{ Hr}}{\text{DAY}} \times \frac{365 \text{ DAYS}}{\text{YR}} \times \frac{15 \text{ YRS}}{1} \times \frac{\text{hr}}{3600 \text{ S}} = 1314 \text{ hr}$$

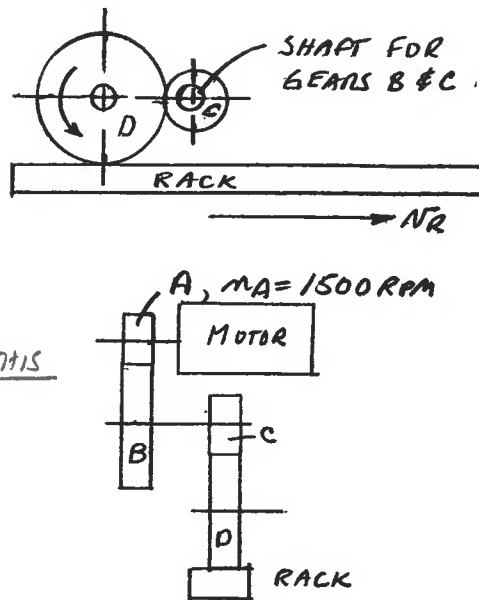
USE 1500 hr DESIGN LIFE.

SUMMARY: POWER = 5.0 hp, $K_o = 1.50$, $P_{\text{DES}} = 7.5 \text{ hp}$, QUALITY = $A_V = 11$

	P_d	N_p	N_g	D_p	D_g	F	C	
PAIR 1:	10	17	172	1.700	14.800	1.200	8.250	WELL BALANCED IN SIZE.
PAIR 2:	6	16	73	2.667	12.167	2.500	7.417	

ALL GEARS MADE FROM SAE 4340: PAIR 1 - OQT 1100; HB 321

PAIR 2 - OQT 900; HB 388



DESIGN OF SPUR GEARS	
APPLICATION: [Problem 78 - First pair Rack and pinion drive driven by a fluid power motor]	
Double reduction - First pair - Input Data: Overload Factor: $K_o = 1.50$ Table 9-7 Transmitted Power: $P = 5$ hp Design Power $P_{des} = 7.5$ hp Diametral Pitch: $P_d = 10$ Fig. 9-24 Input Speed: $n_P = 1500$ rpm Number of Pinion Teeth: $N_P = 17$ Desired Output Speed: $n_G = 172$ rpm Computed number of gear teeth: 148.3 Enter: Chosen No. of Gear Teeth: $N_G = 148$	
Computed data: Actual Output Speed: $n_G = 172.30$ rpm Gear Ratio: $m_G = 8.71$ Pitch Diameter - Pinion: $D_P = 1.700$ in Pitch Diameter - Gear: $D_G = 14.800$ in Center Distance: $C = 8.250$ in Pitch Line Speed: $v_t = 668$ ft/min Transmitted Load: $W_t = 247$ lb	
Secondary Input Data: Face Width Guidelines (in): Min 0.800 Nom 1.200 Max 1.600 Enter: Face Width: $F = 1.200$ in Ratio: Face width/pinion diameter: $F/D_P = 0.71$ Recommended range of ratio: $0.50 < F/D_P < 2.00$ Enter: Elastic Coefficient: $C_p = 2300$ Table 9-10 Enter: Quality Number: $A_v = 11$ Table 9-3 Dynamic Factor: $K_v = 1.35$ Table 9-9 [Factors for computing K_v]: $B = 0.826$ $C = 59.75$ Reference: $N_P = 17$ $N_G = 148$ Bending Geometry Factor-Pinion: $J_P = 0.295$ Fig. 9-15 Bending Geometry Factor-Gear: $J_G = 0.425$ Fig. 9-15 Reference: $m_G = 8.71$ Enter: Pitting Geometry Factor: $I = 0.110$ Fig. 9-21	
Factors in Design Analysis: Alignment Factor, $K_m = 1.0 + C_{pf} + C_{ma}$ If $F < 1.0$ 0.046 If $F > 1.0$ 0.048 $[0.50 < F/D_P < 2.00]$ Pinion Proportion Factor, $C_{pf} = 0.046$ Figure 9-16 Enter: $C_{pf} = 0.048$ Type of gearing: Open Commer. Precision Ex. Prec. Mesh Alignment Factor, $C_{ma} = 0.267$ 0.146 0.083 0.050 Enter: $C_{ma} = 0.146$ Figure 9-17 Alignment Factor: $K_m = 1.19$ [Computed]	
Size Factor: $K_s = 1.00$ Table 9-8: Use 1.00 if $P_d \geq 5$ Pinion Rim Thickness Factor: $K_{BP} = 1.00$ Fig. 9-18: Use 1.00 if solid blank Gear Rim Thickness Factor: $K_{BG} = 1.00$ Fig. 9-18: Use 1.00 if solid blank Service Factor: $SF = 1.00$ Use 1.00 if no unusual conditions Reliability Factor: $K_R = 1.00$ Table 9-11 Use 1.00 for $R = .99$	
Enter: Design Life: 1500 hours See Table 9-12 Pinion - Number of load cycles: $N_P = 1.4E+08$ Guidelines: Y_N, Z_N Gear - Number of load cycles: $N_G = 1.6E+07$ 10^7 cycles $> 10^7$ $< 10^7$ Bending Stress Cycle Factor: $Y_{NP} = 0.97$ 1.00 0.97 Fig. 9-22 Bending Stress Cycle Factor: $Y_{NG} = 1.01$ 1.00 1.01 Fig. 9-22 Pitting Stress Cycle Factor: $Z_{NP} = 0.94$ 1.00 0.94 Fig. 9-23 Pitting Stress Cycle Factor: $Z_{NG} = 0.99$ 1.00 0.99 Fig. 9-23	
Stress Analysis: Bending Pinion: Required $s_{at} = 17,347$ psi See Fig. 9-11 or Gear: Required $s_{at} = 11,564$ psi Table 9-5	
Stress Analysis: Pitting Pinion: Required $s_{ac} = 126,062$ psi See Fig. 9-12 or Gear: Required $s_{ac} = 119,695$ psi Table 9-5	
Required hardness of pinion HB: 301 Equations in Fig. 9-12-Grade 1 Required hardness of gear HB: 281 Equations in Fig. 9-12-Grade 1 Specify materials, alloy and heat treatment, for most severe requirement.	
One possible material specification: Pinion requires HB 301; SAE 4340 OQT 1100; HB 321; 19% elongation Gear requires HB 281; SAE 4340 OQT 1100; HB 321; 19% elongation	
Comments: Same material used for pinion and gear because contact stresses are close.	

DESIGN OF SPUR GEARS			
APPLICATION: [Problem 78 - Second pair Rack and pinion drive driven by a fluid power motor]			
Double reduction - Second pair - Input Data:			
Overload Factor: $K_o =$	1.50	Table 9-7	
Transmitted Power: $P =$	5 hp		
Design Power $P_{des} =$	7.5 hp		
Diametral Pitch: $P_d =$	6	Fig. 9-24	
Input Speed: $n_P =$	172.3 rpm		
Number of Pinion Teeth: $N_P =$	16		
Desired Output Speed: $n_G =$	38.2 rpm		
Computed number of gear teeth:	72.2		
Enter: Chosen No. of Gear Teeth: $N_G =$	73		
Computed data:			
Actual Output Speed: $n_G =$	37.76 rpm		
Gear Ratio: $m_G =$	4.56		
Pitch Diameter - Pinion: $D_P =$	2.667 in		
Pitch Diameter - Gear: $D_G =$	12.167 in		
Center Distance: $C =$	7.417 in		
Pitch Line Speed: $v_t =$	120 ft/min		
Transmitted Load: $W_t =$	1372 lb		
Secondary Input Data:			
Face Width Guidelines (in):	Min 1.333	Nom 2.000	Max 2.667
Enter: Face Width: $F =$	2.500		
Ratio: Face width/pinion diameter: $F/D_P =$	0.94		
Recommended range of ratio: $0.50 < F/D_P < 2.00$			
Enter: Elastic Coefficient: $C_p =$	2300	Table 9-10	
Enter: Quality Number: $A_v =$	11	Table 9-3	
Dynamic Factor: $K_v =$	1.15	Table 9-9	
[Factors for computing K_v]: $B =$	0.826	$C = 59.75$	
Reference: $N_P = 16$	$N_G = 73$		
Bending Geometry Factor-Pinion: $J_P =$	0.265	Fig. 9-15	
Bending Geometry Factor-Gear: $J_G =$	0.400	Fig. 9-15	
Reference: $m_G =$	4.56		
Enter: Pitting Geometry Factor: $I =$	0.102	Fig. 9-21	
Factors in Design Analysis:			
Alignment Factor, $K_m = 1.0 + C_{pf} + C_{ma}$	If $F < 1.0$	If $F > 1.0$	
Pinion Proportion Factor, $C_{pf} =$	0.069	0.088	$[0.50 < F/D_P < 2.00]$
Enter: $C_{pf} =$	0.088	Figure 9-16	
Type of gearing: Open	Commer.	Precision	Ex. Prec.
Mesh Alignment Factor, $C_{ma} =$	0.288	0.166	0.099
Enter: $C_{ma} =$	0.166	Figure 9-17	
Alignment Factor: $K_m =$	1.25	[Computed]	
Size Factor: $K_s =$	1.00	Table 9-8: Use 1.00 if $P_d \geq 5$	
Pinion Rim Thickness Factor: $K_{BP} =$	1.00	Fig. 9-18: Use 1.00 if solid blank	
Gear Rim Thickness Factor: $K_{BG} =$	1.00	Fig. 9-18: Use 1.00 if solid blank	
Service Factor: $SF =$	1.00	Use 1.00 if no unusual conditions	
Reliability Factor: $K_R =$	1.00	Table 9-11 Use 1.00 for $R = .99$	
Enter: Design Life: 1500 hours		See Table 9-12	
Pinion - Number of load cycles: $N_P = 1.6E+07$		Guidelines: Y_N, Z_N	
Gear - Number of load cycles: $N_G = 3.4E+06$	10 ⁷ cycles	$> 10^7$	$< 10^7$
Bending Stress Cycle Factor: $Y_{NP} =$	1.01	1.00	1.01
Bending Stress Cycle Factor: $Y_{NG} =$	1.04	1.00	1.04
Pitting Stress Cycle Factor: $Z_{NP} =$	0.99	1.00	0.99
Pitting Stress Cycle Factor: $Z_{NG} =$	1.03	1.00	1.03
Stress Analysis: Bending			
Pinion: Required $s_{at} =$	26,592 psi		See Fig. 9-11 or Table 9-5
Gear: Required $s_{at} =$	17,109 psi		
Stress Analysis: Pitting			
Pinion: Required $s_{ac} =$	153,423 psi		See Fig. 9-12 or Table 9-5
Gear: Required $s_{ac} =$	147,465 psi		
Required hardness of pinion HB:	386	Equations in Fig. 9-12-Grade 1	
Required hardness of gear HB:	368	Equations in Fig. 9-12-Grade 1	
Specify materials, alloy and heat treatment, for most severe requirement.			
One possible material specification:			
Pinion requires HB 386; SAE 4340 OQT 900; HB 388; 14% elongation			
Gear requires HB 368; SAE 4340 OQT 900; HB 388; 14% elongation			
Comments:			
Same material and heat treatment used for pinion and gear.			
Same material used for both Pair 1 and Pair 2; Different tempering temperatures.			

79.

GEAR DRIVE FOR A LIFT TRUCK
ROLLING WHEEL IS THE INVERSE OF
A PINION DRIVING A RACK.

N AT CENTER OF WHEEL EQUALS
SPEED OF LIFT TRUCK.

$$N = \pi D_w m_w / 12$$

$$m_w = \frac{12 N}{\pi D_w} = \frac{12 (1760 \text{ FT/MIN})}{\pi (12 \text{ IN})} = 560.23 \text{ RPM}$$

$$\text{TRAIN VALUE} = 3000 / 560.23 = 5.355$$

THIS TV COULD BE PRODUCED BY ONE PAIR OF GEARS.
HOWEVER, FOR POWER REQUIRED, GEAR WOULD BE TOO
LARGE TO ATTACH TO AXLE WITH 12-IN WHEEL.
USE DOUBLE REDUCTION. SEE TWO FOLLOWING PAGES.

$$VR_1 = \frac{N_B}{N_A} = \frac{50}{17} = 2.94$$

$$VR_2 = \frac{N_D}{N_C} = \frac{38}{21} = 1.81$$

$$m_w = m_D = m_A \cdot \frac{N_A N_C}{N_B N_D}$$

$$m_w = 3000 \cdot \frac{17}{50} \cdot \frac{21}{38} = 563.7 \text{ RPM}$$

SLIGHTLY HIGHER
THAN TARGET.

NOTES GEARS HAVE $P_d = 6$ FOR PAIR 1

AND $P_d = 5$ FOR PAIR 2. FACE WIDTHS ARE
RELATIVELY LARGE. $F_1 = 2.50 \text{ IN}$, $F_2 = 3.00 \text{ IN}$.

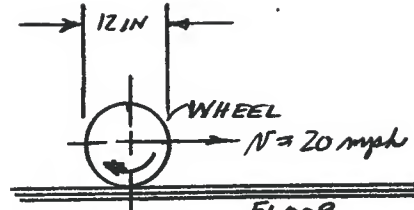
REDESIGN WITH HELICAL GEARS SHOULD ALLOW
A SMALLER SYSTEM.

ALSO ALL GEARS USE SAME MATERIAL AND
HEAT TREATMENT.

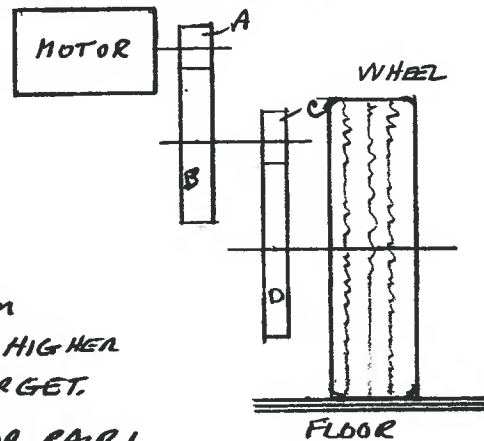
LIFE CALCULATION:

$$\frac{16 \text{ h}}{\text{DAY}} \times \frac{6 \text{ DAYS}}{\text{WK}} \times \frac{52 \text{ WKS}}{\text{YR}} \times 20 \text{ YR} = 99840 \text{ h} \quad \text{USE } L = 100000 \text{ h}$$

DESIGN DECISIONS: SF = 1.00, KR = 1.50 [1 FAILURE IN 10000; R = 0.9999]



$$N = \frac{20 \text{ MI}}{\text{hr}} \times \frac{5280 \text{ FT}}{\text{MI}} \times \frac{1 \text{ hr}}{60 \text{ MIN}} = 1760 \text{ FT/MIN.}$$



CONTINUED ON NEXT TWO PAGES

DESIGN OF SPUR GEARS			
APPLICATION: Problem 79 - First pair Industrial lift truck drive to wheels: driven by DC motor			
Double reduction - First pair - Input Data:			
Overload Factor:	$K_o = 1.50$	Table 9-7	
Transmitted Power:	$P = 20$	hp	
Design Power P_{des} :	$P_{des} = 30$	hp	
Diametral Pitch:	$P_d = 6$	Fig. 9-24	
Input Speed:	$n_P = 3000$	rpm	
Number of Pinion Teeth:	$N_P = 17$		
Desired Output Speed:	$n_G = 1000$	rpm	
Computed number of gear teeth:	51.0		
Enter: Chosen No. of Gear Teeth:	$N_G = 50$		
Computed data:			
Actual Output Speed:	$n_G = 1020.00$	rpm	
Gear Ratio:	$m_G = 2.94$		
Pitch Diameter - Pinion:	$D_P = 2.833$	in	
Pitch Diameter - Gear:	$D_G = 8.333$	in	
Center Distance:	$C = 5.583$	in	
Pitch Line Speed:	$v_t = 2225$	ft/min	
Transmitted Load:	$W_t = 297$	lb	
Secondary Input Data:			
Face Width Guidelines (in):	Min 1.333	Nom 2.000	Max 2.667
Enter: Face Width:	$F = 2.500$	in	
Ratio: Face width/pinion diameter:	$F/D_P = 0.88$		
Recommended range of ratio:	$0.50 < F/D_P < 2.00$		
Enter: Elastic Coefficient:	$C_p = 2300$	Table 9-10	
Enter: Quality Number:	$A_v = 7$	Table 9-3	
Dynamic Factor:	$K_v = 1.19$	Table 9-9	
[Factors for computing K_v]	$B = 0.397$	$C = 83.77$	
Reference:	$N_P = 17$	$N_G = 50$	
Bending Geometry Factor-Pinion:	$J_P = 0.293$	Fig. 9-15	
Bending Geometry Factor-Gear:	$J_G = 0.380$	Fig. 9-15	
Reference:	$m_G = 2.94$		
Enter: Pitting Geometry Factor:	$I = 0.097$	Fig. 9-21	
Factors in Design Analysis:			
Alignment Factor, $K_m = 1.0 + C_{pt} + C_{ma}$	If $F < 1.0$	If $F > 1.0$	
Pinion Proportion Factor, C_{pt}	$C_{pt} = 0.063$	0.082	$[0.50 < F/D_P < 2.00]$
Enter: C_{pt}	$C_{pt} = 0.082$	Figure 9-16	
Type of gearing:	Open	Commer.	Precision
Mesh Alignment Factor, C_{ma}	$C_{ma} = 0.288$	0.166	0.099
Enter: C_{ma}	$C_{ma} = 0.166$	Figure 9-17	
Alignment Factor: K_m	$K_m = 1.25$	[Computed]	
Size Factor: K_s	$K_s = 1.00$	Table 9-8: Use 1.00 if $P_d \geq 5$	
Pinion Rim Thickness Factor: K_{RP}	$K_{RP} = 1.00$	Fig. 9-18: Use 1.00 if solid blank	
Gear Rim Thickness Factor: K_{RG}	$K_{RG} = 1.00$	Fig. 9-18: Use 1.00 if solid blank	
Service Factor: SF	$SF = 1.00$	Use 1.00 if no unusual conditions	
Reliability Factor: K_R	$K_R = 1.50$	Table 9-11 Use 1.00 for $R = .99$	
Enter: Design Life:	100000	hours	See Table 9-12
Pinion - Number of load cycles: N_P	$N_P = 1.8E+10$	Guidelines: Y_N, Z_N	
Gear - Number of load cycles: N_G	$N_G = 6.1E+09$	10^7 cycles	$> 10^7$
Bending Stress Cycle Factor: Y_{NP}	$Y_{NP} = 0.89$	1.00	Fig. 9-22
Bending Stress Cycle Factor: Y_{NG}	$Y_{NG} = 0.91$	1.00	Fig. 9-22
Pitting Stress Cycle Factor: Z_{NP}	$Z_{NP} = 0.84$	1.00	Fig. 9-23
Pitting Stress Cycle Factor: Z_{NG}	$Z_{NG} = 0.86$	1.00	Fig. 9-23
Stress Analysis: Bending			
Pinion: Required s_{at}	$s_{at} = 9,152$	psi	See Fig. 9-11 or Table 9-5
Gear: Required s_{at}	$s_{at} = 6,901$	psi	
Stress Analysis: Pitting			
Pinion: Required s_{ac}	$s_{ac} = 127,576$	psi	See Fig. 9-12 or Table 9-5
Gear: Required s_{ac}	$s_{ac} = 124,609$	psi	
Required hardness of pinion HB:	306	Equations in Fig. 9-12-Grade 1	
Required hardness of gear HB:	297	Equations in Fig. 9-12-Grade 1	
Specify materials, alloy and heat treatment, for most severe requirement.			
One possible material specification:			
Pinion requires HB 306; SAE 4340 OQT 1100; HB 321; 19% elongation			
Gear requires HB 297; SAE 4340 OQT 1100; HB 321; 19% elongation			
Comments:			
Same material used for pinion and gear because contact stresses are close.			

DESIGN OF SPUR GEARS			
APPLICATION: [Problem 79 - Second pair Industrial lift truck drive to wheels; driven by DC motor]			
Double reduction - Second pair - Input Data:			
Overload Factor:	$K_o = 1.50$	Table 9-7	
Transmitted Power:	$P = 20$	hp	
Design Power P_{des} :	$P_{des} = 30$	hp	
Diametral Pitch:	$P_d = 5$	Fig. 9-24	
Input Speed:	$n_P = 1020$	rpm	
Number of Pinion Teeth:	$N_P = 21$		
Desired Output Speed:	$n_G = 560.23$	rpm	
Computed number of gear teeth:	38.2		
Enter: Chosen No. of Gear Teeth:	$N_G = 38$		
Computed data:			
Actual Output Speed:	$n_G = 563.68$	rpm	
Gear Ratio:	$m_G = 1.81$		
Pitch Diameter - Pinion:	$D_P = 4.200$	in	
Pitch Diameter - Gear:	$D_G = 7.600$	in	
Center Distance:	$C = 5.900$	in	
Pitch Line Speed:	$v_t = 1122$	ft/min	
Transmitted Load:	$W_t = 598$	lb	
Secondary Input Data:			
Face Width Guidelines (in):	Min 1.600	Nom 2.400	Max 3.200
Enter: Face Width:	$F = 3.000$	in	
Ratio: Face width/pinion diameter: $F/D_P = 0.71$			
Recommended range of ratio: $0.50 < F/D_P < 2.00$			
Enter: Elastic Coefficient:	$C_p = 2300$	Table 9-10	
Enter: Quality Number:	$A_v = 7$	Table 9-3	
Dynamic Factor:	$K_v = 1.14$	Table 9-9	
[Factors for computing K_v]: $B = 0.397$	$C = 83.77$		
Reference:	$N_P = 21$	$N_G = 38$	
Bending Geometry Factor-Pinion:	$J_P = 0.330$	Fig. 9-15	
Bending Geometry Factor-Gear:	$J_G = 0.380$	Fig. 9-15	
Reference:	$m_G = 1.81$		
Enter: Pitting Geometry Factor:	$I = 0.092$	Fig. 9-21	
Factors In Design Analysis:			
Alignment Factor, $K_m = 1.0 + C_{pf} + C_{ma}$	If $F < 1.0$	If $F > 1.0$	
Pinion Proportion Factor, $C_{pf} = 0.046$	0.071	$[0.50 < F/D_P < 2.00]$	
Enter: $C_{pf} = 0.071$	Figure 9-16		
Type of gearing: Open	Commer.	Precision	Ex. Prec.
Mesh Alignment Factor, $C_{ma} = 0.296$	0.173	0.105	0.068
Enter: $C_{ma} = 0.173$	Figure 9-17		
Alignment Factor: $K_m = 1.24$	[Computed]		
Size Factor: $K_s = 1.00$	Table 9-8: Use 1.00 if $P_d \geq 5$		
Pinion Rim Thickness Factor: $K_{BP} = 1.00$	Fig. 9-18: Use 1.00 if solid blank		
Gear Rim Thickness Factor: $K_{BG} = 1.00$	Fig. 9-18: Use 1.00 if solid blank		
Service Factor: $SF = 1.00$	Use 1.00 if no unusual conditions		
Reliability Factor: $K_R = 1.50$	Table 9-11 Use 1.00 for $R = .99$		
Enter: Design Life: 100000 hours	See Table 9-12		
Pinion - Number of load cycles: $N_P = 6.1E+09$	Guidelines: Y_N, Z_N		
Gear - Number of load cycles: $N_G = 3.4E+09$	10 ⁷ cycles	$> 10^7$	$< 10^7$
Bending Stress Cycle Factor: $Y_{NP} = 0.91$	1.00	0.91	Fig. 9-22
Bending Stress Cycle Factor: $Y_{NG} = 0.92$	1.00	0.92	Fig. 9-22
Pitting Stress Cycle Factor: $Z_{NP} = 0.86$	1.00	0.86	Fig. 9-23
Pitting Stress Cycle Factor: $Z_{NG} = 0.87$	1.00	0.87	Fig. 9-23
Stress Analysis: Bending			
Pinion: Required $s_{at} = 10,447$ psi			See Fig. 9-11 or
Gear: Required $s_{at} = 8,974$ psi			Table 9-5
Stress Analysis: Pitting			
Pinion: Required $s_{ac} = 131,992$ psi			See Fig. 9-12 or
Gear: Required $s_{ac} = 130,475$ psi			Table 9-5
Required hardness of pinion HB:	320	Equations in Fig. 9-12-Grade 1	
Required hardness of gear HB:	315	Equations in Fig. 9-12-Grade 1	
Specify materials, alloy and heat treatment, for most severe requirement.			
One possible material specification:			
Pinion requires HB 320; SAE 4340 OQT 1100; HB 321; 19% elongation			
Gear requires HB 315; SAE 4340 OQT 1100; HB 388; 19% elongation			
Comments:			
Same material and heat treatment used for pinion and gear.			
Same material used for both Pair 1 and Pair 2; Same tempering temperature.			

DESIGN OF PLASTIC SPUR GEARS			
Application:			
Small band saw driven by an electric motor			
Problem 80 - One possible design			
Initial Input Data:			
Input Power:	$P =$	0.5 hp	
Input Speed:	$n_P =$	860 rpm	
Diametral Pitch:	$P_d =$	12	
Number of Pinion Teeth:	$N_P =$	18	
Desired Output Speed:	$n_G =$	266 rpm	
Computed number of gear teeth:		58.195	
Enter: Chosen No. of Gear Teeth:	$N_G =$	58	
Computed data:			
Actual Output Speed:	$n_G =$	266.9 rpm	
Gear Ratio:	$m_G =$	3.222	
Pitch Diameter - Pinion:	$D_P =$	1.500 in	
Pitch Diameter - Gear:	$D_G =$	4.833 in	
Center Distance:	$C =$	6.333 in	
Pitch Line Speed:	$v_t =$	337.7 ft/min	
Transmitted Load:	$W_t =$	48.84 lb	
Secondary Input Data - Pinion:			
Tooth Form:	20 degree full depth		
Lewis Form Factor:	$Y =$	0.521	Table 9-15
Safety Factor:	$SF =$	1.50	Ref. Table 9-7
Material:	Nylon		
Allowable Bending Stress:	$s_{at} =$	6000 psi	Table 9-14 or Fig. 9-28
Required Face Width:	$F =$	0.281 in	
Enter: Specified Face Width:	$F =$	0.300 in	
Actual Bending Stress in Pinion:	$s_t =$	5624 psi	
Secondary Input Data - Gear:			
Tooth Form:	20 degree full depth	Same as for pinion	
Lewis Form Factor:	$Y =$	0.709	Table 9-15
Safety Factor:	$SF =$	1.50	Same as for pinion
Material:	Acetal		
Allowable Bending Stress:	$s_{at} =$	5000 psi	Table 9-14 or Fig. 9-28
Face Width - Gear:	$F =$	0.300 in	Same as for pinion
Actual Bending Stress in Gear:	$s_t =$	4133 psi	Must be $< s_{at}$

DESIGN OF PLASTIC SPUR GEARS**Application:**

Paper feed roll driven by an electric motor

Problem 81 - One possible design

Initial Input Data:Input Power: $P = 0.06$ hpInput Speed: $n_P = 88$ rpmDiametral Pitch: $P_d = 20$ Number of Pinion Teeth: $N_P = 16$ Desired Output Speed: $n_G = 21$ rpm

Computed number of gear teeth: 67.048

Enter: Chosen No. of Gear Teeth: $N_G = 67$ **Computed data:**Actual Output Speed: $n_G = 21.0$ rpmGear Ratio: $m_G = 4.188$ Pitch Diameter - Pinion: $D_P = 0.800$ inPitch Diameter - Gear: $D_G = 3.350$ inCenter Distance: $C = 4.150$ inPitch Line Speed: $v_t = 18.4$ ft/minTransmitted Load: $W_t = 107.4$ lb**Secondary Input Data - Pinion:**

Tooth Form: 20 degree stub

Lewis Form Factor: $Y = 0.578$

Table 9-15

Safety Factor: $SF = 1.25$

Ref. Table 9-7

Material: Nylon-glass filled

Allowable Bending Stress: $s_{at} = 12000$ psi

Table 9-14 or Fig. 9-28

Required Face Width: $F = 0.387$ inEnter: Specified Face Width: $F = 0.400$ inActual Bending Stress in Pinion: $s_t = 11612$ psi**Secondary Input Data - Gear:**

Tooth Form: 20 degree stub

Same as for pinion

Lewis Form Factor: $Y = 0.782$

Table 9-15

Safety Factor: $SF = 1.25$

Same as for pinion

Material: Nylon-glass filled

Allowable Bending Stress: $s_{at} = 12000$ psi

Table 9-14 or Fig. 9-28

Face Width - Gear: $F = 0.400$ in

Same as for pinion

Actual Bending Stress in Gear: $s_t = 8583$ psiMust be $< s_{at}$

DESIGN OF PLASTIC SPUR GEARS			
Application:			
Wheels of remote control car-Electric motor drive			
Problem 82 - One possible design			
Initial Input Data:			
Input Power:	$P =$	0.025 hp	
Input Speed:	$n_P =$	430 rpm	
Diametral Pitch:	$P_d =$	48	
Number of Pinion Teeth:	$N_P =$	14	
Desired Output Speed:	$n_G =$	121 rpm	
Computed number of gear teeth:		49.752	
Enter: Chosen No. of Gear Teeth:	$N_G =$	50	
Computed data:			
Actual Output Speed:	$n_G =$	120.4 rpm	
Gear Ratio:	$m_G =$	3.571	
Pitch Diameter - Pinion:	$D_P =$	0.292 in	
Pitch Diameter - Gear:	$D_G =$	1.042 in	
Center Distance:	$C =$	1.333 in	
Pitch Line Speed:	$v_t =$	32.8 ft/min	
Transmitted Load:	$W_t =$	25.1 lb	
Secondary Input Data - Pinion:			
Tooth Form:	20 degree stub		
Lewis Form Factor:	$Y =$	0.54	Table 9-15
Safety Factor:	$SF =$	1.25	Ref: Table 9-7
Material:	Nylon-unfilled		
Allowable Bending Stress:	$s_{at} =$	6000 psi	Table 9-14 or Fig. 9-28
Required Face Width:	$F =$	0.465 in	
Enter: Specified Face Width:	$F =$	0.470 in	
Actual Bending Stress in Pinion:	$s_t =$	5938 psi	
Secondary Input Data - Gear:			
Tooth Form:	20 degree stub		Same as for pinion
Lewis Form Factor:	$Y =$	0.758	Table 9-15
Safety Factor:	$SF =$	1.25	Same as for pinion
Material:	Nylon-unfilled		
Allowable Bending Stress:	$s_{at} =$	6000 psi	Table 9-14 or Fig. 9-28
Face Width - Gear:	$F =$	0.470 in	Same as for pinion
Actual Bending Stress in Gear:	$s_t =$	4230 psi	Must be $< s_{at}$

DESIGN OF PLASTIC SPUR GEARS**Application:**

Food-chopping machine driven by electric motor

Problem 83 - One possible design

Initial Input Data:Input Power: $P = 0.65$ hpInput Speed: $n_P = 1560$ rpmDiametral Pitch: $P_d = 10$ Number of Pinion Teeth: $N_P = 18$ Desired Output Speed: $n_G = 469$ rpm

Computed number of gear teeth: 59.872

Enter: Chosen No. of Gear Teeth: $N_G = 60$ **Computed data:**Actual Output Speed: $n_G = 468.0$ rpmGear Ratio: $m_G = 3.333$ Pitch Diameter - Pinion: $D_P = 1.125$ inPitch Diameter - Gear: $D_G = 3.750$ inCenter Distance: $C = 4.875$ inPitch Line Speed: $v_t = 459.5$ ft/minTransmitted Load: $W_t = 46.7$ lb**Secondary Input Data - Pinion:**

Tooth Form: 20 degree full depth

Lewis Form Factor: $Y = 0.521$

Table 9-15

Safety Factor: $SF = 1.75$

Ref. Table 9-7

Material: Nylon-unfilled

Allowable Bending Stress: $s_{at} = 6000$ psi

Table 9-14 or Fig. 9-28

Required Face Width: $F = 0.418$ inEnter: Specified Face Width: $F = 0.420$ inActual Bending Stress in Pinion: $s_t = 5971$ psi**Secondary Input Data - Gear:**

Tooth Form: 20 degree full depth Same as for pinion

Lewis Form Factor: $Y = 0.713$

Table 9-15

Safety Factor: $SF = 1.75$

Same as for pinion

Material: Acetal-unfilled

Allowable Bending Stress: $s_{at} = 5000$ psi

Table 9-14 or Fig. 9-28

Face Width - Gear: $F = 0.420$ in

Same as for pinion

Actual Bending Stress in Gear: $s_t = 4363$ psiMust be $< s_{at}$

CHAPTER 10

HELICAL GEARS, BEVEL GEARS, AND WORMGEARING

1. HELICAL GEARS: $P_d = 8$, $\phi_t = 14\frac{1}{2}^\circ$, $N_g = 45$, $F = 2.01$ IN, $\psi = 30^\circ$
- a. $P = 5.0$ HP, $M_G = 1250$ RPM; $TORQUE = T = \frac{63000(5.0)}{1250} = 252$ LB·IN
- $$W_t = \frac{T}{D_g/2} : D_g = \frac{N_g}{P_d} = \frac{45}{8} = 5.625$$
- $$W_t = \frac{252 \text{ LB·IN}}{5.625/2} = 89.6 \text{ LB}$$
- $$W_x = W_t \tan \psi = 89.6 \tan 30^\circ = 51.7 \text{ LB}$$
- $$W_r = W_t \tan \phi_t = 89.6 \tan 14\frac{1}{2}^\circ = 23.2 \text{ LB}$$
- b. $N_p = 15$; DRIVE TO A RECIRCULATING PUMP, $K_o = 1.50$
- $$S_{tP} = \frac{W_t P_d}{F J_p} \cdot \frac{K_o K_s K_m K_B K_v}{\text{ASSUME } K_s = K_B = 1.0}$$
- APPROXIMATE J_p FROM FIG 10-5. DATA ARE FOR $\phi_m = 15^\circ$
- $$\text{ACTUAL } \phi_m = \tan^{-1}[\tan \phi_t \cos \psi] = \tan^{-1}[\tan 14\frac{1}{2}^\circ \cos 30^\circ] = 12.6^\circ$$
- $$J_p \approx (0.38)(0.97) = 0.369$$
- $$D_p = N_p/P_d = 15/8 = 1.875$$
- ;
- $F/D_p = 1.067$
- ;
- $C_{pf} = 0.075$
- ;
- $C_{md} = 1.55$
- $$K_m = 1.0 + C_{pf} + C_{md} = 1.06 + 0.075 + 0.155 = 1.23$$
- $$V_s = \pi D_g M_G/12 = \pi(5.625)(1250)/12 = 1841 \text{ FT/MIN}$$
- SPECIFY $A_v = 11$; THEN $K_v = 1.55$ (FIG 9-20)
- $$S_{tP} = \frac{(89.6)(8)}{(2.00)(0.369)} (1.50)(1.0)(1.23)(1.0)(1.55) = 2778 \text{ PSI}$$
- $$S_c = C_p \sqrt{\frac{W_t K_o K_B K_m K_v}{F D_p I}} = 1960 \sqrt{\frac{(89.6)(1.50)(1.0)(1.23)(1.55)}{(2.00)(1.875)(0.20)}}$$
- $$S_c = 3622 \text{ PSI} : \text{I EST. FROM TABLE 10-1.}$$
- c. SPECIFIED CAST IRON BECAUSE OF LOW STRESSES
- GRAY CAST IRON: $S_{at} = 5000 \text{ PSI}$, $S_{ac} = 5000 \text{ PSI}$
- CLASS 20

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HELICAL GEARS: $P = 2.50 \text{ HP}$, $N_P = 16$, $N_G = 48$, $P_d = 12$, $\phi_n = 20^\circ$, $\psi = 45^\circ$

$$F = 1.50 \text{ IN: TORQUE} = T = \frac{63000(2.50)}{1750} = 90.0 \text{ LB-IN}$$

$$M_G = 1750 \text{ RPM: } W_t = \frac{T}{D_G/2} = \frac{90.0}{(5.657/2)} = 31.8 \text{ LB}$$

FROM PROBLEM 8-42: $P_d = 8.485$; $D_G = 5.657 \text{ IN}$, $\phi_t = 27.2^\circ$

$$D_P = \frac{N_P}{P_d} = \frac{16}{8.485} = 1.887 \text{ IN}$$

$$W_x = W_t \tan \psi = 31.8 \text{ LB} \cdot \tan 45^\circ = 31.8 \text{ LB}$$

$$W_L = W_t \tan \phi_t = 31.8 \cdot \tan 27.2^\circ = 16.4 \text{ LB}$$

b. $K_o = 1.25$ (LT. SHOCK); $K_s = K_R = 1.00$; $J_P = 0.30$

$$N_G = \pi D_G M_G / 12 = \pi (5.657)(1750) / 12 = 2592 \text{ FT/MIN}$$

CENTRIFUGAL BLOWER LET $A_v = 9$; $K_v = 1.40$

$$F/D_P = 1.50/1.887 = 0.795 \quad C_{pf} = 0.05, C_{md} = 0.15; K_m = 1.20$$

$$S_t = \frac{W_t P_d}{F J_P} K_o K_s K_m K_R K_v = \frac{(31.8)(8.485)(1.25)(1.0)(1.2)(1.40)}{(1.50)(0.30)} = 1259 \text{ PSI}$$

PITTING: $I = 0.21$, $C_P = 1960$ CAST IRON

$$S_c = C_P \sqrt{\frac{W_t K_o K_s K_m K_R}{F D_P I}} = 1960 \sqrt{\frac{(31.8)(1.25)(1.0)(1.2)(1.40)}{(1.50)(1.887)(0.21)}} = 20,775 \text{ PSI}$$

c. $S_c = 20,775 \text{ PSI}$: SPECIFY CLASS 20 CAST IRON
 $S_{at} = 5000 \text{ PSI}$, $S_{ac} = 5000 \text{ PSI}$

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HELICAL GEARS: $P = 15 \text{ HP}$, $N_P = 12$, $N_G = 36$, $P_d = 6$, $\phi_t = 14\frac{1}{2}^\circ$, $\psi = 45^\circ$, $F = 1.00 \text{ IN}$

$$M_G = 2200 \text{ RPM; } T = \frac{63000 P}{n} = \frac{63000(15)}{2200} = 430 \text{ LB-IN}$$

FROM PROB. 8-43: $D_G = 6.00 \text{ IN}$; $D_P = \frac{N_P}{P_d} = \frac{12}{6} = 2.00 \text{ IN}$

a. $W_t = T / (D_G/2) = 430 / (6.00/2) = 143 \text{ LB}$

$$W_x = W_t \tan \psi = 143 \text{ LB} \cdot \tan 45^\circ = 143 \text{ LB}$$

$$W_L = W_t \tan \phi_t = 143 \text{ LB} \cdot \tan 14\frac{1}{2}^\circ = 37.0 \text{ LB}$$

$K_o = 2.00$ CONC. MIXER
 $K_s = 1.00 = K_R$

$$\frac{F}{D_P} = \frac{1.00}{2.00} = 0.50; C_{pf} = 0.025$$

$$C_{md} = 0.15; K_m = 1.18$$

b. $N_G = \pi D_G M_G / 12 = \pi (6.00)(2200) / 12 = 3456 \text{ FT/MIN}$

$A_v = 9$; $K_v = 1.44$; $J_P = 0.30$ (EST.), $I = 0.190$ (EST)

$$S_t = \frac{W_t P_d}{F J_P} K_o K_s K_m K_R K_v = \frac{(143)(6)(1.0)(1.18)(1.0)(1.44)(2.0)}{(1.00)(0.30)} = 9720 \text{ PSI}$$

USE NODULAR (DUCTILE) IRON $C_P = 2050$

$$S_c = C_P \sqrt{\frac{W_t K_o K_s K_m K_R}{F D_P I}} = 2050 \sqrt{\frac{(143)(2.0)(1.0)(1.18)(1.44)}{(1.0)(2.00)(0.19)}} = 73300 \text{ PSI}$$

c. SPECIFY: DUCTILE IRON ASTM A536 60-40-18 OR C5 CLASS 40
 $S_{at} = 22000 \text{ PSI}$, $S_{ac} = 77000 \text{ PSI}$ $S_{at} = 13 \text{ KSI}$, $S_{ac} = 75 \text{ KSI}$

NOTE: PROB. 8-43 GIVES $P_x = \text{AXIAL PITCH} = 0.5236 \text{ IN}$

THEN $F/P_x = 1.00/0.5236 = 1.91$ - LOW - SHOULD BE 72.0 FOR FULL HELIX ACTION.

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HELICAL GEARS: $P = 0.50 \text{ hp}$, $M_G = 3450 \text{ RPM}$, $P_m = 24$, $\phi_m = 14\frac{1}{2}^\circ$
 $\psi = 45^\circ$, $N_G = 72$, $N_P = 16$, $F = 0.25 \text{ IN}$, WINCH-MOD. SHOCK $K_0 = 1.50$
 FROM PROB B-44: $P_d = 16.97$, $D_G = 4.243 \text{ IN}$ $D_P = \frac{N_P}{P_d} = \frac{16}{16.97} = 0.943 \text{ IN}$

$$T = \frac{63000(P)}{M_G} = \frac{63000(0.50)}{3450} = 9.13 \text{ LB-IN}$$

$$a. \quad W_t = T / (D_G/2) = (9.13) / (4.243/2) = 4.30 \text{ LB}$$

$$W_x = W_t \tan \psi = 4.30 \text{ LB} \cdot \tan 45^\circ = 4.30 \text{ LB}$$

$$W_n = W_t \tan \phi_c = 4.30 \tan 20^\circ = 1.57 \text{ LB}; \phi_c = 20.0^\circ \text{ FROM PROB B-44.}$$

$$b. \quad N_c = \pi D_G M_G / 12 = \pi (4.243)(3450) / 12 = 3832 \text{ FT/MIN}$$

$$\text{LET } A_N = 9, K_N = 1.48, K_S = 1.00 = K_B, J_P = 0.32 \text{ (EST)} \quad I = 0.22 \text{ (EST)}$$

$$F/D_P = 0.25/0.943 = 0.265; C_P \approx 0; C_m = 0.14; K_m = 1.14$$

$$S_t = \frac{W_t P_d}{F J} \cdot K_0 K_S K_m K_B K_N = \frac{(4.30)(16.97)(1.50)(1.00)(1.14)(1.48)}{(0.25)(0.32)}$$

$$S_t = 2308 \text{ PSI}$$

TRY CAST IRON

$$S_c = C_P \sqrt{\frac{W_t K_0 K_S K_m K_B K_N}{F D_P I}} = 1960 \sqrt{\frac{(4.30)(1.50)(1.00)(1.14)(1.48)}{(0.25)(0.943)(0.25)}}$$

$$S_c = 26633 \text{ PSI}$$

$$\text{SPECIFY CLASS 20 CAST IRON } S_{AT} = 5000 \text{ PSI } S_{CL} = 5000 \text{ PSI}$$

NOTE: PROBLEM B-44 GIVES $P_x = \text{AXIAL PITCH} = 0.1815 \text{ IN}$
 THEN $F/P_x = 0.25 \text{ IN} / 0.1815 \text{ IN} = 1.35$ - LOW - SHOULD BE 72.0
 FOR FULL HELICAL ACTION.

THE FOLLOWING PAGES GIVE SAMPLE DESIGNS FOR PROBLEMS 5-11. THE PROCEDURE IS SIMILAR TO THAT USED IN EXAMPLE PROBLEM 10-2. OTHER DESIGNS ARE POSSIBLE. READER IS ENCOURAGED TO WORK TOWARD A PARTICULAR GOAL OF MATERIAL TYPE, CENTER DISTANCE, OVERALL SIZE OR OTHER APPLICATION-SPECIFIC GOAL.

NOTE THAT TRANSVERSE DIAMETRAL PITCH MUST BE INPUT. IF NORMAL DIAMETRAL PITCH IS ORIGINALLY SPECIFIED, COMPUTE $P_d = P_n \cos \psi$.

DESIGN OF HELICAL GEARS-U.S.		APPLICATION:	Reciprocating compressor driven by an electric motor Problem 10-5	
Initial Input Data:			Factors in Design Analysis:	
Input Power:	$P = 5$ hp		Alignment Factor, $K_m = 1.0 + C_{pf} + C_{ma}$	If $F < 1.0$ 0.095 If $F > 1.0$ 0.098
Input Speed:	$n_P = 1200$ rpm		Pinion Proportion Factor, $C_{pf} =$	[0.50 < $F/D_P < 2.00$]
Transverse Diametral Pitch, P_d :	$P_d = 18$		Enter: $C_{pf} = 0.098$	Figure 9-16
Number of Pinion Teeth:	$N_P = 18$		Type of gearing:	Open Commer. Precision Ex. Prec.
Desired Output Speed:	$n_G = 387.5$ rpm		Mesh Alignment Factor, $C_{ma} =$	0.267 0.146 0.083 0.050
Computed number of gear teeth:	55.7		Enter: $C_{ma} = 0.146$	Figure 9-17
Enter: Chosen No. of Gear Teeth:	$N_G = 56$		Alignment Factor: $K_m =$	1.24 [Computed]
Computed data:				
Actual Output Speed:	$n_G = 385.7$ rpm		Overload Factor: $K_o =$	1.50 Table 9-7
Gear Ratio:	$m_G = 3.11$		Size Factor: $K_s =$	1.00 Table 9-8: Use 1.00 if $P_d \geq 5$
Pitch Diameter - Pinion:	$D_P = 1.000$ in		Pinion Rim Thickness Factor: $K_{RP} =$	1.00 Fig. 9-18: Use 1.00 if solid blank
Pitch Diameter - Gear:	$D_G = 3.111$ in		Gear Rim Thickness Factor: $K_{RG} =$	1.00 Fig. 9-18: Use 1.00 if solid blank
Center Distance:	$C = 2.056$ in		Dynamic Factor: $K_v =$	1.24 [Computed: See Fig. 9-20]
Pitch Line Speed:	$V_t = 314$ ft/min		Service Factor: $SF =$	1.00 Use 1.00 if no unusual conditions
Transmitted Load:	$W_t = 525$ lb		Reliability Factor: $K_R =$	1.00 Table 9-11 Use 1.00 for $R = .99$
Secondary Input Data:			Enter: Design Life: 15000 hours	
Transverse pressure angle:	$\phi_t = 20.0$ deg		Pinion - Number of load cycles: $N_P =$	1.1E+09 See Table 9-12
Helix angle:	$\psi = 25.0$ deg		Gear - Number of load cycles: $N_G =$	3.5E+08
Axial Pitch:	$p_x = 0.3743$ in		Bending Stress Cycle Factor: $Y_{NP} =$	0.94 Fig. 9-22
Min. Face Width (2 x Axial Pitch):	$F_{min} = 0.749$ in		Bending Stress Cycle Factor: $Y_{NG} =$	0.96 Fig. 9-22
Enter: Face Width:	$F = 1.200$ in		Pitting Stress Cycle Factor: $Z_{NP} =$	0.90 Fig. 9-24
Enter: Elastic Coefficient:	$C_p = 2300$	Table 9-10	Pitting Stress Cycle Factor: $Z_{NG} =$	0.92 Fig. 9-24
Enter: Quality Number:	$A_v = 11$	Table 9-3	Stress Analysis: Bending	
REF: $N_P, N_G =$	18 56		Pinion: Required $s_{at} =$	42,786 psi See Fig. 9-11 or Table 9-5
Enter: Bending Geometry Factors:			Gear: Required $s_{at} =$	39,050 psi
Pinion:	$J_P = 0.453$	Fig 10-5,6,7	Pinion: Required $s_{ec} =$	179,572 psi See Fig. 9-12 or Table 9-5
Gear:	$J_G = 0.486$	Fig 10-5,6,7	Gear: Required $s_{ec} =$	175,668 psi
Enter: Pitting Geometry Factor:	$I = 0.205$	Tab. 10-1,2	Stress Analysis: Pitting	
REF: $m_G =$	3.11		Specify materials, alloy and heat treatment, for most severe requirement.	
One possible material specification: Steel pinion and gear: Carburized, Grade 1				
Axial Force:	$W_x = 245$ lb		Pinion requires HRC 58 min.: SAE 4320 SOQT 450; HRC 59; Carburized	
Radial Force:	$W_r = 191$ lb		Gear requires HRC 58 min.: SAE 4320 SOQT 450; HRC 59; Carburized	

DESIGN OF HELICAL GEARS-U.S.		APPLICATION:		Milling machine driven by an electric motor Problem 10-6	
Initial Input Data:				Factors in Design Analysis:	
Input Power:	$P = 20$ hp	Pinion Proportion Factor, $C_{pf} = 1.0 + C_{pf} + C_{ma}$		If $F < 1.0$	If $F > 1.0$
Input Speed:	$n_P = 550$ rpm			0.079	0.098
Transverse Diametral Pitch, P_d :	$P_d = 10$	Enter: $C_{pf} = 0.098$		Figure 9-16	
Number of Pinion Teeth:	$N_P = 24$	Type of gearing:		Open	Commer.
Desired Output Speed:	$n_G = 185$ rpm	Mesh Alignment Factor, $C_{ma} = 0.288$		0.166	0.099
Computed number of gear teeth:	71.4	Enter: $C_{ma} = 0.099$		Figure 9-17	
Enter: Chosen No. of Gear Teeth:	$N_G = 72$	Alignment Factor: $K_m = 1.20$		[Computed]	
Computed data:				Overload Factor: $K_o = 1.50$	
Actual Output Speed:	$n_G = 183.3$ rpm	Size Factor: $K_s = 1.00$		Table 9-7	
Gear Ratio:	$m_G = 3.00$	Pinion Rim Thickness Factor: $K_{BP} = 1.00$		Table 9-8: Use 1.00 if $P_d \geq 5$	
Pitch Diameter - Pinion:	$D_P = 2.400$ in	Gear Rim Thickness Factor: $K_{BG} = 1.00$		Fig. 9-18: Use 1.00 if solid blank	
Pitch Diameter - Gear:	$D_G = 7.200$ in	Dynamic Factor: $K_v = 1.16$		Fig. 9-18: Use 1.00 if solid blank	
Center Distance:	$C = 4.800$ in	Service Factor: $SF = 1.00$		[Computed: See Fig. 9-20]	
Pitch Line Speed:	$V_t = 346$ ft/min	Reliability Factor: $K_R = 1.25$		Use 1.00 if no unusual conditions	
Transmitted Load:	$W_t = 1910$ lb	Enter: Design Life: 15000 hours		Table 9-11 Use 1.00 for $R = .99$	
Secondary Input Data:				Guidelines: Y_N, Z_N	
Transverse pressure angle:	$\phi_t = 20.0$ deg	Pinion - Number of load cycles: $N_P = 5.0E+08$		10 ⁷ cycles >10 ⁷ <10 ⁷	
Helix angle:	$\psi = 15.0$ deg	Gear - Number of load cycles: $N_G = 1.7E+08$		1.00 0.95 Fig. 9-22	
Axial Pitch:	$p_x = 1.1725$ in	Bending Stress Cycle Factor: $Y_{NP} = 0.95$		1.00 0.97 Fig. 9-22	
Min. Face Width (2 x Axial Pitch):	$F_{min} = 2.345$ in	Bending Stress Cycle Factor: $Y_{NG} = 0.97$		1.00 0.91 Fig. 9-24	
Enter: Face Width:	$F = 2.500$ in	Pitting Stress Cycle Factor: $Z_{NP} = 0.91$		1.00 0.94 Fig. 9-24	
Enter: Elastic Coefficient:	$C_p = 2300$	Pitting Stress Cycle Factor: $Z_{NG} = 0.94$			
Enter: Quality Number:	$A_v = 9$	Stress Analysis: Bending			
REF: $N_P, N_G = 24$	72	Pinion: Required $s_{at} = 43,560$ psi		See Fig. 9-11 or Table 9-5	
		Gear: Required $s_{at} = 38,931$ psi			
Enter: Bending Geometry Factors:				Stress Analysis: Pitting	
Pinion:	$J_P = 0.480$	Pinion: Required $s_{ac} = 173,320$ psi		See Fig. 9-12 or Table 9-5	
Gear:	$J_G = 0.526$	Gear: Required $s_{ac} = 167,788$ psi			
Enter: Pitting Geometry Factor:	$I = 0.220$	Specify materials, alloy and heat treatment, for most severe requirement.			
REF: $m_G = 3.00$		One possible material specification: Steel pinion and gear: Carburized, Grade 1			
Axial Force:	$W_x = 512$ lb	Pinion requires HRC 58 min.: SAE 4320 SOQT 450; HRC 59; Carburized			
Radial Force:	$W_r = 695$ lb	Gear requires HRC 58 min.: SAE 4320 SOQT 450; HRC 59; Carburized			

DESIGN OF HELICAL GEARS-U.S.		APPLICATION:		Punch press driven by an electric motor Problem 10-7	
Initial Input Data:				Factors in Design Analysis:	
Input Power: $P = 50$ hp		Pinion Proportion Factor, $C_{pf} = 0.038$		Alignment Factor, $K_m = 1.0 + C_{pf} + C_{ma}$	
Input Speed: $n_P = 900$ rpm		Enter: $C_{pf} = 0.056$		Figure 9-16	
Transverse Diametral Pitch, $P_d = 6$		Type of gearing: Open		Commer. Precision Ex. Prec.	
Number of Pinion Teeth: $N_P = 24$		Mesh Alignment Factor, $C_{ma} = 0.288$		0.166 0.099 0.063	
Desired Output Speed: $n_G = 227.5$ rpm		Enter: $C_{ma} = 0.166$		Figure 9-17	
Computed number of gear teeth: 94.9		Alignment Factor: $K_m = 1.22$		[Computed]	
Enter: Chosen No. of Gear Teeth: $N_G = 95$		Overload Factor: $K_o = 1.75$		Table 9-7	
Computed data:		Size Factor: $K_s = 1.00$		Table 9-8: Use 1.00 if $P_d \geq 5$	
Actual Output Speed: $n_G = 227.4$ rpm		Pinion Rim Thickness Factor: $K_{ap} = 1.00$		Fig. 9-18: Use 1.00 if solid blank	
Gear Ratio: $m_G = 3.96$		Gear Rim Thickness Factor: $K_{ag} = 1.00$		Fig. 9-18: Use 1.00 if solid blank	
Pitch Diameter - Pinion: $D_P = 4.000$ in		Dynamic Factor: $K_v = 1.26$		[Computed: See Fig. 9-20]	
Pitch Diameter - Gear: $D_G = 15.833$ in		Service Factor: $SF = 1.00$		Use 1.00 if no unusual conditions	
Center Distance: $C = 9.917$ in		Reliability Factor: $K_R = 1.25$		Table 9-11 Use 1.00 for $R = .99$	
Pitch Line Speed: $V_t = 942$ ft/min		Enter: Design Life: 15000 hours		See Table 9-12	
Transmitted Load: $W_t = 1751$ lb		Pinion - Number of load cycles: $N_P = 8.1E+08$		Guidelines: Y_N, Z_N	
Secondary Input Data:		Gear - Number of load cycles: $N_G = 2.0E+08$		10 ⁷ cycles >10 ⁷ <10 ⁷	
Transverse pressure angle: $\phi_t = 20.0$ deg		Bending Stress Cycle Factor: $Y_{NP} = 0.94$		1.00 0.94 Fig. 9-22	
Helix angle: $\psi = 25.0$ deg		Bending Stress Cycle Factor: $Y_{NG} = 0.96$		1.00 0.96 Fig. 9-22	
Axial Pitch: $p_x = 1.1229$ in		Pitting Stress Cycle Factor: $Z_{NP} = 0.90$		1.00 0.90 Fig. 9-24	
Min. Face Width (2 x Axial Pitch): $F_{min} = 2.246$ in		Pitting Stress Cycle Factor: $Z_{NG} = 0.93$		1.00 0.93 Fig. 9-24	
Enter: Face Width: $F = 2.500$ in		Stress Analysis: Bending			
Enter: Elastic Coefficient: $C_p = 2300$		Pinion: Required $s_{at} = 32,253$ psi		See Fig. 9-11 or Table 9-5	
Enter: Quality Number: $A_v = 9$		Gear: Required $s_{at} = 29,607$ psi			
REF: $m_G = 24$ 95		Stress Analysis: Pitting			
Enter: Bending Geometry Factors:		Pinion: Required $s_{ac} = 147,637$ psi		See Fig. 9-12 or Table 9-5	
Pinion: $J_P = 0.465$ Fig 10-5,6,7		Gear: Required $s_{ac} = 142,875$ psi			
Gear: $J_G = 0.496$ Fig 10-5,6,7					
Enter: Pitting Geometry Factor: $I = 0.220$ Tab. 10-1,2					
REF: $m_G = 3.96$					
Axial Force: $W_x = 816$ lb					
Radial Force: $W_r = 637$ lb					
Specify materials, alloy and heat treatment, for most severe requirement.					
One possible material specification: Steel pinion and gear: Through hardened					
Pinion requires HB 368: SAE 4140 OQT 900; HB 388					
Gear requires HB 353: SAE 4140 OQT 900; HB 388					

DESIGN OF HELICAL GEARS-U.S.		APPLICATION:	Small cement mixer driven by a gasoline engine Problem 10-8		Use steel pinion with cast iron gear	
Initial Input Data:			Factors in Design Analysis:			
Input Power:	$P = 2.5$ hp		Alignment Factor, $K_m = 1.0 + C_{pf} + C_{ma}$		If $F < 1.0$ 0.080 If $F > 1.0$ 0.089 $F/D_p = 1.05$	
Input Speed:	$n_p = 900$ rpm		Pinion Proportion Factor, $C_{pf} =$		[0.50 < F/D_p < 2.00]	
Transverse Diametral Pitch, P_d :	$P_d = 12$		Enter: $C_{pf} = 0.089$		Figure 9-16	
Number of Pinion Teeth:	$N_p = 20$		Type of gearing:		Open Commer. Precision Ex. Prec.	
Desired Output Speed:	$n_g = 75$ rpm		Mesh Alignment Factor, $C_{ma} = 0.276$		0.154 0.090 0.056	
Computed number of gear teeth:	240.0		Enter: $C_{ma} = 0.276$		Figure 9-17	
Enter: Chosen No. of Gear Teeth:	$N_g = 240$		Alignment Factor: $K_m = 1.37$		[Computed]	
Computed data:			Overload Factor: $K_o = 2.00$ Table 9-7			
Actual Output Speed:	$n_g = 75.0$ rpm		Size Factor: $K_s = 1.00$ Table 9-8: Use 1.00 if $P_d \geq 5$			
Gear Ratio:	$m_g = 12.00$		Pinion Rim Thickness Factor: $K_{BP} = 1.00$ Fig. 9-18: Use 1.00 if solid blank			
Pitch Diameter - Pinion:	$D_p = 1.667$ in		Gear Rim Thickness Factor: $K_{BG} = 1.00$ Fig. 9-18: Use 1.00 if solid blank			
Pitch Diameter - Gear:	$D_g = 20.000$ in		Dynamic Factor: $K_v = 1.33$ [Computed: See Fig. 9-20]			
Center Distance:	$C = 10.833$ in		Service Factor: $SF = 1.00$ Use 1.00 if no unusual conditions			
Pitch Line Speed:	$v_t = 393$ ft/min		Reliability Factor: $K_R = 1.00$ Table 9-11 Use 1.00 for $R = .99$			
Transmitted Load:	$W_t = 210$ lb		Enter: Design Life: 8000 hours See Table 9-12			
Secondary Input Data:			Pinion - Number of load cycles: $N_p = 4.3E+08$			
Transverse pressure angle:	$\phi_t = 20.0$ deg		Gear - Number of load cycles: $N_g = 3.6E+07$			
Helix angle:	$\psi = 25.0$ deg		Bending Stress Cycle Factor: $Y_{NP} = 0.95$ 1.00 0.95 Fig. 9-22			
Axial Pitch:	$p_x = 0.5614$ in		Bending Stress Cycle Factor: $Y_{NG} = 0.99$ 1.00 0.99 Fig. 9-22			
Min. Face Width (2 x Axial Pitch):	$F_{min} = 1.123$ in		Pitting Stress Cycle Factor: $Z_{NP} = 0.92$ 1.00 0.92 Fig. 9-24			
Enter: Face Width:	$F = 1.750$ in		Pitting Stress Cycle Factor: $Z_{NG} = 0.97$ 1.00 0.97 Fig. 9-24			
Enter: Elastic Coefficient:	$C_p = 2100$ Table 9-10		Stress Analysis: Bending			
Enter: Quality Number:	$A_v = 12$ Table 9-3		Pinion: Required $s_{at} = 12,180$ psi See Fig. 9-11 or Table 9-5			
REF: $m_g, N_g = 20$ 240			Gear: Required $s_{at} = 10,295$ psi			
Enter: Bending Geometry Factors:			Stress Analysis: Pitting			
Pinion:	$J_p = 0.451$ Fig 10-5,6,7		Pinion: Required $s_{ec} = 72,310$ psi See Fig. 9-12 or Table 9-5			
Gear:	$J_g = 0.512$ Fig 10-5,6,7		Gear: Required $s_{ec} = 68,583$ psi			
Enter: Pitting Geometry Factor:	$I = 0.260$ Tab. 10-1,2		Specify materials, alloy and heat treatment, for most severe requirement.			
REF: $m_g = 12.00$			One possible material specification: Steel pinion and cast iron gear			
Axial Force:	$W_x = 98$ lb		Pinion requires HB 134: SAE 1020 CD; HB 160 - Or almost any steel			
Radial Force:	$W_r = 76$ lb		Gear: Grade 40 gray cast iron; HB 201; $s_{at} = 13$ ksi; $s_{ec} = 75$ ksi (Table 9-6)			

DESIGN OF HELICAL GEARS-U.S.		APPLICATION:		Wood chipper driven by a gasoline engine Problem 10-9		Speed increaser - cells changed	
Initial Input Data:				Factors in Design Analysis:			
Input Power: $P = 75$ hp		Pinion Speed: $n_p = 2200$ rpm		Alignment Factor, $K_m = 1.0 + C_{pf} + C_{ma}$		If $F < 1.0$ 0.044 0.054 If $F > 1.0$ 0.054 $F/D_p = 0.69$ [0.50 < F/D_p < 2.00]	
Transverse Diametral Pitch, $P_d = 10$		Number of Pinion Teeth: $N_p = 26$		Enter: $C_{pf} = 0.054$ Figure 9-16		Type of gearing: Open Commer. Precision Ex. Prec.	
Desired Output Speed: $n_p = 4550$ rpm		Computed number of gear teeth: $N_g = 53.8$		Mesh Alignment Factor, $C_{ma} = 0.277$ 0.155 0.090 0.056		Enter: $C_{me} = 0.277$ Figure 9-17	
Enter: Chosen No. of Gear Teeth: $N_g = 54$				Alignment Factor: $K_m = 1.33$ [Computed]			
Computed data:				Overload Factor: $K_o = 2.75$ Table 9-7			
Actual Output Speed: $n_g = 4569.2$ rpm		Gear Ratio: $m_g = 2.08$		Size Factor: $K_s = 1.00$ Table 9-8: Use 1.00 if $P_d \geq 5$			
Pitch Diameter - Pinion: $D_p = 2.600$ in		Pitch Diameter - Gear: $D_g = 5.400$ in		Pinion Rim Thickness Factor: $K_{ap} = 1.00$ Fig. 9-18: Use 1.00 if solid blank			
Center Distance: $C = 4.000$ in		Pitch Line Speed: $v_t = 3097$ ft/min		Gear Rim Thickness Factor: $K_{ag} = 1.00$ Fig. 9-18: Use 1.00 if solid blank			
Transmitted Load: $W_t = 799$ lb				Dynamic Factor: $K_v = 1.44$ [Computed: See Fig. 9-20]			
				Service Factor: $SF = 1.00$ Use 1.00 if no unusual conditions			
				Reliability Factor: $K_R = 1.00$ Table 9-11 Use 1.00 for $R = .99$			
Secondary Input Data:				Enter: Design Life: 8000 hours See Table 9-12			
Transverse pressure angle: $\phi_t = 20.0$ deg		Helix angle: $\psi = 25.0$ deg		Pinion - Number of load cycles: $N_p = 1.1E+09$		Guidelines: Y_N, Z_N	
Axial Pitch: $p_x = 0.67372$ in		Min. Face Width (2 x Axial Pitch): $F_{min} = 1.347$ in		Gear - Number of load cycles: $N_g = 5.1E+08$		10 ⁷ cycles > 10 ⁷ < 10 ⁷	
Enter: Face Width: $F = 1.800$ in				Bending Stress Cycle Factor: $Y_{NP} = 0.94$		1.00 0.94 Fig. 9-22	
Enter: Elastic Coefficient: $C_p = 2300$ Table 9-10				Bending Stress Cycle Factor: $Y_{NG} = 0.95$		1.00 0.95 Fig. 9-22	
Enter: Quality Number: $A_v = 9$ Table 9-3				Pitting Stress Cycle Factor: $Z_{NP} = 0.90$		1.00 0.90 Fig. 9-24	
REF: $N_p, N_g = 26$				Pitting Stress Cycle Factor: $Z_{NG} = 0.91$		1.00 0.91 Fig. 9-24	
Enter: Bending Geometry Factors:				Stress Analysis: Bending			
Pinion: $J_p = 0.453$ Fig 10-5,6,7		Gear: $J_g = 0.463$ Fig 10-5,6,7		Pinion: Required $s_{at} = 55,024$ psi		See Fig. 9-11 or Table 9-5	
Enter: Pitting Geometry Factor: $I = 0.188$ Tab. 10-1,2		REF: $m_g = 2.08$		Gear: Required $s_{at} = 53,268$ psi			
Axial Force: $W_x = 373$ lb		Radial Force: $W_r = 291$ lb		Stress Analysis: Pitting			
				Pinion: Required $s_{ac} = 176,932$ psi		See Fig. 9-12 or Table 9-5	
				Gear: Required $s_{ac} = 174,988$ psi			
Specify materials, alloy and heat treatment, for most severe requirement.				Stress Analysis: Pitting			
One possible material specification: Steel pinion+gear: Carburized-Case hard.							
Pinion requires HRC 55: SAE 4118 DOQT 300; HRC 62							
Gear requires HRC 55: SAE 4118 DOQT 300; HRC 62							

DESIGN OF HELICAL GEARS-U.S.		APPLICATION:		Small tractor driven by a gasoline engine Problem 10-10		Use steel pinion with steel gear	
Initial Input Data:				Factors in Design Analysis:			
Input Power:	$P =$	20 hp		Alignment Factor, $K_m = 1.0 + C_{pf} + C_{ma}$	If $F < 1.0$	If $F > 1.0$	$F/D_p = 0.64$ [0.50 < $F/D_p < 2.00$]
Input Speed:	$n_P =$	450 rpm		Pinion Proportion Factor, $C_{pf} =$	0.039	0.055	
Transverse Diametral Pitch, P_d :	$P_d =$	6		Enter: $C_{pf} =$ 0.055 Figure 9-16			
Number of Pinion Teeth:	$N_P =$	21		Type of gearing:	Open	Commer.	Precision
Desired Output Speed:	$n_G =$	77.5 rpm		Mesh Alignment Factor, $C_{ma} =$	0.284	0.162	0.096
Computed number of gear teeth:		121.9		Enter: $C_{ma} =$ 0.162 Figure 9-17			
Enter: Chosen No. of Gear Teeth:	$N_G =$	122		Alignment Factor: $K_m =$	1.22	[Computed]	
Computed data:				Overload Factor: $K_o =$ 2.75 Table 9-7			
Actual Output Speed:	$n_G =$	77.5 rpm		Size Factor: $K_s =$	1.00	Table 9-8: Use 1.00 if $P_d \geq 5$	
Gear Ratio:	$m_G =$	5.81		Pinion Rim Thickness Factor: $K_{BP} =$	1.00	Fig. 9-18: Use 1.00 if solid blank	
Pitch Diameter - Pinion:	$D_P =$	3.500 in		Gear Rim Thickness Factor: $K_{BG} =$	1.00	Fig. 9-18: Use 1.00 if solid blank	
Pitch Diameter - Gear:	$D_G =$	20.333 in		Dynamic Factor: $K_v =$	1.27	[Computed: See Fig. 9-20]	
Center Distance:	$C =$	11.917 in		Service Factor: $SF =$	1.00	Use 1.00 if no unusual conditions	
Pitch Line Speed:	$V_t =$	412 ft/min		Reliability Factor: $K_R =$	1.00	Table 9-11 Use 1.00 for $R = .99$	
Transmitted Load:	$W_t =$	1801 lb		Enter: Design Life: 8000 hours See Table 9-12			
Secondary Input Data:				Pinion - Number of load cycles: $N_P = 2.2E+08$			
Transverse pressure angle:	$\phi_t =$	20.0 deg		Gear - Number of load cycles: $N_G = 3.7E+07$	Guidelines: Y_N, Z_N		
Helix angle:	$\psi =$	25.0 deg		10 ⁷ cycles >10 ⁷ <10 ⁷			
Axial Pitch:	$p_x =$	1.1229 in		Bending Stress Cycle Factor: $Y_{NP} =$	0.96	1.00	Fig. 9-22
Min. Face Width (2 x Axial Pitch):	$F_{min} =$	2.246 in		Bending Stress Cycle Factor: $Y_{NG} =$	0.99	1.00	Fig. 9-22
Enter: Face Width:	$F =$	2.250 in		Pitting Stress Cycle Factor: $Z_{NP} =$	0.93	1.00	Fig. 9-24
Enter: Elastic Coefficient:	$C_p =$	2300	Table 9-10	Pitting Stress Cycle Factor: $Z_{NG} =$	0.97	1.00	Fig. 9-24
Enter: Quality Number:	$A_v =$	11	Table 9-3	Stress Analysis: Bending			
REF: $N_P, N_G =$	21	122		Pinion: Required $s_{at} =$	42,676 psi	See Fig. 9-11 or Table 9-5	
Enter: Bending Geometry Factors:				Gear: Required $s_{at} =$	37,194 psi		
Pinion:	$J_P =$	0.444	Fig 10-5,6,7	Stress Analysis: Pitting			
Gear:	$J_G =$	0.494	Fig 10-5,6,7	Pinion: Required $s_{ac} =$	148,576 psi	See Fig. 9-12 or Table 9-5	
Enter: Pitting Geometry Factor:	$I =$	0.240	Tab. 10-1,2	Gear: Required $s_{ac} =$	142,449 psi		
REF: $m_G =$	5.81			Specify materials, alloy and heat treatment, for most severe requirement.			
One possible material specification: Steel pinion and gear. Through hardened							
Axial Force:	$W_x =$	746 lb		Pinion requires HB 371: SAE 4340 OQT 900; HB 388			
Radial Force:	$W_r =$	583 lb		Gear requires HB 352: SAE 4340 OQT 1000; HB 363			

DESIGN OF HELICAL GEARS-U.S.		APPLICATION:		Electric power generator driven by a water turbine Problem 10-11		Use steel pinion with steel gear	
Initial Input Data:				Factors in Design Analysis:			
Input Power: $P = 15$ hp		Pinion Proportion Factor, $C_{pf} = 0.053$		If $F < 1.0$		If $F > 1.0$	
Input Speed: $n_P = 4500$ rpm		Pinion Proportion Factor, $C_{pf} = 0.053$		0.050		0.053	
Transverse Diametral Pitch, $P_d = 12$		Enter: $C_{pf} = 0.053$		Figure 9-16		Figure 9-16	
Number of Pinion Teeth: $N_P = 20$		Type of gearing: Open		Commer.		Precision	
Desired Output Speed: $n_G = 3600$ rpm		Mesh Alignment Factor, $C_{ma} = 0.268$		0.147		0.083	
Computed number of gear teeth: $N_G = 25$		Enter: $C_{ma} = 0.147$		Figure 9-17		Figure 9-17	
Enter: Chosen No. of Gear Teeth: $N_G = 25$		Alignment Factor: $K_m = 1.20$		[Computed]		[Computed]	
Computed data:				Overload Factor: $K_o = 1.20$			
Actual Output Speed: $n_G = 3600.0$ rpm				Size Factor: $K_s = 1.00$			
Gear Ratio: $m_G = 1.25$				Table 9-8: Use 1.00 if $P_d \geq 5$			
Pitch Diameter - Pinion: $D_P = 1.667$ in				Fig. 9-18: Use 1.00 if solid blank			
Pitch Diameter - Gear: $D_G = 2.083$ in				Fig. 9-18: Use 1.00 if solid blank			
Center Distance: $C = 1.875$ in				Dynamic Factor: $K_v = 1.36$			
Pitch Line Speed: $V_t = 1963$ ft/min				[Computed: See Fig. 9-20]			
Transmitted Load: $W_t = 252$ lb				Service Factor: $SF = 1.00$			
				Use 1.00 if no unusual conditions			
				Reliability Factor: $K_R = 1.25$			
				Table 9-11 Use 1.00 for $R = .99$			
Secondary Input Data:				Enter: Design Life: 100000 hours			
Transverse pressure angle: $\phi_t = 20.0$ deg				See Table 9-12			
Helix angle: $\psi = 25.0$ deg				Guidelines: Y_N, Z_N			
Axial Pitch: $p_x = 0.5614$ in				10 ⁷ cycles $> 10^7$			
Min. Face Width (2 x Axial Pitch): $F_{min} = 1.123$ in				1.00 0.88 Fig. 9-22			
Enter: Face Width: $F = 1.250$ in				1.00 0.89 Fig. 9-22			
Enter: Elastic Coefficient: $C_p = 2300$				1.00 0.83 Fig. 9-24			
Enter: Quality Number: $A_v = 9$				1.00 0.84 Fig. 9-24			
REF: $N_P, N_G = 20, 25$							
Enter: Bending Geometry Factors:				Stress Analysis: Bending			
Pinion: $J_P = 0.418$ Fig 10-5,6,7				Pinion: Required $s_{at} = 16,093$ psi			
Gear: $J_G = 0.432$ Fig 10-5,6,7				Gear: Required $s_{at} = 15,397$ psi			
Enter: Pitting Geometry Factor: $I = 0.150$ Tab. 10-1,2				Stress Analysis: Pitting			
REF: $m_G = 1.25$				Pinion: Required $s_{ec} = 137,623$ psi			
Axial Force: $W_x = 118$ lb				Gear: Required $s_{ec} = 135,985$ psi			
Radial Force: $W_r = 92$ lb				Specify materials, alloy and heat treatment, for most severe requirement.			
				One possible material specification: Steel pinion and gear: Through hardened			
				Pinion requires HB 337: SAE 4340 OQT 1000; HB 363			
				Gear requires HB 332: SAE 4340 OQT 1000; HB 363			

HELICAL GEARS		APPLICATION:	
POWER TRANSMISSION CAPACITY		Centrifugal pump driven by an electric motor	
Chapter 10-Problem 12		Used: $A_v = 12$; $L = 15000$ h	
Initial Input Data: Enter: Face Width: $F = 2.500$ in Input Speed: $n_p = 1725$ rpm Diametral Pitch: $P_d = 9.659$ Number of Pinion Teeth: $N_p = 20$ Number of Gear Teeth: $N_g = 75$		Factors in Design Analysis: Alignment Factor, $K_m = 1.0 + C_{pt} + C_{ma}$ If $F < 1.0$ If $F > 1.0$ If $F/D_p < 2.00$ If $F/D_p = 1.21$ Pinion Proportion Factor, $C_{pt} = 0.096$ 0.114 0.114 Figure 9-16 Enter: $C_{pt} = 0.114$ Type of gearing: Open Commer. Precision Ex. Prec. Mesh Alignment Factor, $C_{ma} = 0.288$ 0.166 0.099 0.063 Enter: $C_{ma} = 0.166$ Figure 9-17 Alignment Factor: $K_m = 1.28$ [Computed]	
Computed data: Actual Output Speed: $n_g = 460.0$ rpm Gear Ratio: $m_g = 3.75$ Pitch Diameter - Pinion: $D_p = 2.071$ in Pitch Diameter - Gear: $D_g = 7.765$ in Center Distance: $C = 4.918$ in Pitch Line Speed: $v_l = 935$ ft/min Transmitted Load at P_{min} Capacity: $W_l = 797$ lb		Overload Factor: $K_o = 1.25$ Table 9-7 Size Factor: $K_s = 1.00$ Table 9-8: Use 1.00 if $P_d \geq 5$ Pinion Rim Thickness Factor: $K_{ap} = 1.00$ Fig. 9-18: Use 1.00 if solid blank Gear Rim Thickness Factor: $K_{ag} = 1.00$ Fig. 9-18: Use 1.00 if solid blank Dynamic Factor: $K_v = 1.50$ [Computed: See Fig. 9-20] Service Factor: $SF = 1.00$ Use 1.00 if no unusual conditions Reliability Factor: $K_R = 1.25$ Table 9-9 Use 1.00 for $R = .99$	
Power Transmission Capacity: (Using Eq. 9-32, 9-34) Pinion: Based on Bending Stress: 41.43 hp Gear: Based on Bending Stress: 47.41 hp Pinion: Based on Contact Stress: 22.59 hp Gear: Based on Contact Stress: 24.14 hp Power Transmission Capacity: 22.59 hp		Pinion - Number of load cycles: $N_p = 1.6E+09$ See Table 9-7 Gear - Number of load cycles: $N_g = 4.1E+08$ Guidelines: Y_N, Z_N Bending Stress Cycle Factor: $Y_{NP} = 0.93$ 1.00 0.93 Fig. 9-20 Bending Stress Cycle Factor: $Y_{NG} = 0.95$ 1.00 0.95 Fig. 9-20 Pitting Stress Cycle Factor: $Z_{NP} = 0.89$ 1.00 0.89 Fig. 9-22 Pitting Stress Cycle Factor: $Z_{NG} = 0.92$ 1.00 0.92 Fig. 9-22	
Enter: Elastic Coefficient: $C_p = 2300$ Table 9-10 Enter: Quality Number: $A_v = 12$ Table 9-3 REF: $N_p, N_g = 20$ 75 Enter: Bending Geometry Factors: Press. angle = 20 deg Pinion: $J_p = 0.465$ Fig. 10-5, 6, 7 Gear: $J_g = 0.521$ Fig. 10-5, 6, 7 Enter: Pitting Geometry Factor: $I = 0.200$ Tables 10-1, 2 REF: $m_g = 3.75$		Allowable Bending Stress Numbers: (Input) Pinion: $s_{at} = 39,200$ psi See Fig. 9-11 or Table 9-5 Gear: $s_{at} = 39,200$ psi Allowable Contact Stress Numbers: (Input) Pinion: $s_{ac} = 138,900$ psi See Fig. 9-12 or Table 9-5 Gear: $s_{ac} = 138,900$ psi	
Material specification: Steel pinion; Steel gear, Through HT Pinion material: SAE 4140 OQT 1000 341 HB Gear material: SAE 4140 OQT 1000 341 HB		Through-Hardened Grade 1 Steel 39.2 ksi Fig. 9-11 39.2 ksi Fig. 9-11 138.9 ksi Fig. 9-12 138.9 ksi Fig. 9-12	

For K_v :
 B 0.915
 C 54.74

HELICAL GEARS		APPLICATION:	
POWER TRANSMISSION CAPACITY		Centrifugal pump driven by an electric motor	
Chapter 10-Problem 13		Used: $A_v = 12$; $L = 15000$ h	
Initial Input Data:			
Enter: Face Width: $F = 2.500$ in		Enter: $A_v = 12$; $L = 15000$ h	
Input Speed: $n_p = 1725$ rpm			
Diametral Pitch: $P_d = 9.659$			
Number of Pinion Teeth: $N_p = 20$			
Number of Gear Teeth: $N_g = 75$			
Computed data:			
Actual Output Speed: $n_g = 460.0$ rpm			
Gear Ratio: $m_g = 3.75$			
Pitch Diameter - Pinion: $D_p = 2.071$ in			
Pitch Diameter - Gear: $D_g = 7.765$ in			
Center Distance: $C = 4.918$ in			
Pitch Line Speed: $v_t = 935$ ft/min			
Transmitted Load at P_{min} Capacity: $W_t = 1338$ lb			
Power Transmission Capacity: (Using Eq. 9-32, 9-34)			
Pinion: Based on Bending Stress: 58.12 hp			
Gear: Based on Bending Stress: 66.52 hp			
Pinion: Based on Contact Stress: 37.94 hp			
Gear: Based on Contact Stress: 40.54 hp			
Power Transmission Capacity: 37.94 hp			
Enter: Elastic Coefficient: $C_p = 2300$		Table 9-10	
Enter: Quality Number: $A_v = 12$		Table 9-3	
REF: $N_p, N_g = 20, 75$			
Enter: Bending Geometry Factors: Press. angle = 20 deg			
Pinion: $J_p = 0.465$		Fig. 10-5, 6, 7	
Gear: $J_g = 0.521$		Fig. 10-5, 6, 7	
Enter: Pitting Geometry Factor: $I = 0.200$		Tables 10-1, 2	
REF: $m_g = 3.75$			
Factors in Design Analysis:			
Alignment Factor, $K_m = 1.0 + C_{pf} + C_{ma}$		If $F < 1.0$ If $F > 1.0$ $F/D_p = 1.21$	
Pinion Proportion Factor, $C_{pf} = 0.096$		0.114	
Enter: $C_{pf} = 0.114$		Figure 9-16	
Type of gearing: Open		Commer. Precision Ex. Prec.	
Mesh Alignment Factor, $C_{ma} = 0.288$		0.166 0.099 0.063	
Enter: $C_{ma} = 0.166$		Figure 9-17	
Alignment Factor: $K_m = 1.28$		[Computed]	
Overload Factor: $K_o = 1.25$		Table 9-7	
Size Factor: $K_s = 1.00$		Table 9-8: Use 1.00 if $P_d \geq 5$	
Pinion Rim Thickness Factor: $K_{Rp} = 1.00$		Fig. 9-18: Use 1.00 if solid blank	
Gear Rim Thickness Factor: $K_{Rg} = 1.00$		Fig. 9-18: Use 1.00 if solid blank	
Dynamic Factor: $K_v = 1.50$		[Computed: See Fig. 9-20]	
Service Factor: $SF = 1.00$		Use 1.00 if no unusual conditions	
Reliability Factor: $K_R = 1.25$		Table 9-9 Use 1.00 for $R = .99$	
Enter: Design Life: 15000 hours		See Table 9-7	
Pinion - Number of load cycles: $N_p = 1.6E+09$		Guidelines: Y_N, Z_N	
Gear - Number of load cycles: $N_g = 4.1E+08$		10^7 cycles $>10^7$ $<10^7$	
Bending Stress Cycle Factor: $Y_{NP} = 0.93$		1.00 0.93 Fig. 9-20	
Bending Stress Cycle Factor: $Y_{NG} = 0.95$		1.00 0.95 Fig. 9-20	
Pitting Stress Cycle Factor: $Z_{NP} = 0.89$		1.00 0.89 Fig. 9-22	
Pitting Stress Cycle Factor: $Z_{NG} = 0.92$		1.00 0.92 Fig. 9-22	
Allowable Bending Stress Numbers: (Input)			
Pinion: $s_{at} = 55,000$ psi		See Fig. 9-11 or Table 9-5	
Gear: $s_{at} = 55,000$ psi			
Allowable Contact Stress Numbers: (Input)			
Pinion: $s_{ac} = 180,000$ psi		See Fig. 9-12 or Table 9-5	
Gear: $s_{ac} = 180,000$ psi			
Material specification: Steel pinion+gear, Carburized case hard.			
Pinion material: SAE 4620 DOQT 300		62 HRC	
Gear material: SAE 4620 DOQT 300		62 HRC	

For K_v :

B 0.915

C 54.74

Through-Hardened

Grade 1 Steel

17.6 ksi Fig. 9-11

17.6 ksi Fig. 9-11

49.1 ksi Fig. 9-12

49.1 ksi Fig. 9-12

Problem 10-14**BEVEL GEARS****Forces and torque for shaft and bearing load analysis:****On Pinion Shaft - Torque:** $T_P = 630$ lb-inMean radius of pinion: $r_m = 1.052$ in*Enter:* pressure angle: $\phi = 20$ degreesOn Pinion - Tangential load: $W_{tP} = 598.7$ lbOn Pinion - Radial load: $W_{rP} = 206.7$ lbOn Pinion - Axial load: $W_{xP} = 68.9$ lb**On Gear Shaft - Torque:** $T_G = 1890.0$ lb-inOn Gear - Tangential load: $W_{tG} = 598.7$ lbOn Gear - Radial load: $W_{rG} = 68.9$ lbOn Gear - Axial load: $W_{xG} = 206.7$ lb**See following page for stress analysis and design details.**

DESIGN OF BEVEL GEARS		APPLICATION:	
Initial Input Data:			
Input Power:	$P = 3$ hp		
Input Speed:	$n_P = 300$ rpm		
Diametral Pitch:	$P_d = 6$		
Number of Pinion Teeth:	$N_P = 15$		
Desired Output Speed:	$n_G = 100$ rpm		
Computed number of gear teeth:	45.0		
Enter: Chosen No. of Gear Teeth:	$N_G = 45$		
Computed data:			
Actual Output Speed:	$n_G = 100.0$ rpm		
Gear Ratio:	$m_G = 3.00$		
Pitch Diameter - Pinion:	$D_P = 2.500$ in		
Pitch Diameter - Gear:	$D_G = 7.500$ in		
Pitch cone angle - Pinion:	$\gamma = 18.43$ degrees		
Pitch cone angle - Gear:	$\Gamma = 71.57$ degrees		
Outer cone distance:	$A_o = 3.9528$ in		
Pitch Line Speed:	$V_t = 196$ ft/min		
Transmitted Load:	$W_t = 504$ lb		
Secondary Input Data:			
Face Width Guidelines (in):	Nom Max		
Enter: Face Width:	$F = 1.186$ 1.318 1.667		
Enter: Elastic Coefficient:	$C_p = 2300$ Table 9-10		
Enter: Quality Number:	$A_v = 9$ Table 9-3		
Enter: Bending Geometry Factors:			
Pinion:	$J_P = 0.228$ Fig. 10-15		
Gear:	$J_G = 0.190$ Fig. 10-15		
Enter: Pitting Geometry Factor:	$I = 0.078$ Fig. 10-19		
Concrete mixer with moderate shock driven by a gasoline engine			
Problem 10-14 Neither gear straddle mounted			
Load distribution factor, K_m :			
Select from:		From Equation 10-16	
Both gears straddle mounted:	1.00		
One gear straddle mounted:	1.10		
Neither gear straddle mounted:	1.25		
Enter Factor, K_{mb} :	1.25		
$K_m =$	1.26		
Overload Factor: $K_o =$	2.00 Table 9-7	For K_v	
Bending Size Factor: $K_s =$	0.52 Figure 10-13 (for $P_d < 16$)	$B = 0.630$	
Dynamic Factor: $K_v =$	1.121 Computed: Table 9-9	$C = 70.7$	
Pitting Size Factor: $C_s =$	0.59 For $F > 3.14$, $C_s = 0.83$		
For $F < 0.50$, $C_s = 0.5$			
Enter $C_s =$	0.59		
Service Factor: $S_F =$	1.00 Use 1.00 if no unusual conditions		
Bending Reliability Factor: $K_R =$	1.00 Pitting: $C_R =$ 1.00		
Enter: Design Life:	1000 hours See Table 9-7		
Pinion - Number of load cycles: $N_P =$	$1.80E+07$ Y_N, Z_N : Pinion-Fig. 10-16; Gear-Fig. 10-20		
Gear - Number of load cycles: $N_G =$	$6.00E+06$ $10^3 < N < 10^7$ $N > 10^7$		
Bending Stress Cycle Factor: $K_L =$	1.01 2.70 0.84 1.01		
Bending Stress Cycle Factor: $K_L =$	0.96 2.70 0.96 1.03		
Pitting Stress Cycle Factor: $C_L =$	1.27 $N < 10^3$ $N > 10^3$		
Pitting Stress Cycle Factor: $C_L =$	1.36 2.00 1.27		
Stress Analysis - Bending:			
Pinion: Required $s_{at} =$	15,447 psi See Fig. 10-17 or		
Gear: Required $s_{at} =$	19,502 psi Table 10-4		
Stress Analysis - Pitting: Assumes $C_{sc} = 1.5$ for properly crowned teeth			
Pinion: Required $s_{ac} =$	129,992 psi See Fig. 10-21 or		
Gear: Required $s_{ac} =$	121,389 psi Table 10-4		
Specify materials, alloy and heat treatment, for most severe requirement.			
One possible material specification:			
Pinion: HB 312 required: SAE 6150 OQT 1100; HB = 341			
Gear: HB 395 required: SAE 6150 OQT 900; HB = 401			

Problem 10-15**BEVEL GEARS****Forces and torque for shaft and bearing load analysis:****On Pinion Shaft - Torque:** $T_P = 176.4$ lb-inMean radius of pinion: $r_m = 1.093$ in*Enter:* pressure angle: $\phi = 20$ degreesOn Pinion - Tangential load: $W_{tP} = 161.3$ lbOn Pinion - Radial load: $W_{rP} = 52.5$ lbOn Pinion - Axial load: $W_{xP} = 26.3$ lb**On Gear Shaft - Torque:** $T_G = 352.8$ lb-inOn Gear - Tangential load: $W_{tG} = 161.3$ lbOn Gear - Radial load: $W_{rG} = 26.3$ lbOn Gear - Axial load: $W_{xG} = 52.5$ lb**See following page for stress analysis and design details.**

DESIGN OF BEVEL GEARS		APPLICATION:													
Initial Input Data: Input Power: $P = 3.5$ hp Input Speed: $n_P = 1250$ rpm Diametral Pitch: $P_d = 10$ Number of Pinion Teeth: $N_P = 25$ Desired Output Speed: $n_G = 625$ rpm Computed number of gear teeth: Enter: Chosen No. of Gear Teeth: $N_G = 50$		Conveyor with moderate shock driven by a gasoline engine Problem 10-15 Neither gear straddle mounted													
Computed data: Actual Output Speed: $n_G = 625.0$ rpm Gear Ratio: $m_G = 2.00$ Pitch Diameter - Pinion: $D_P = 2.500$ in Pitch Diameter - Gear: $D_G = 5.000$ in Pitch cone angle - Pinion: $\gamma = 26.57$ degrees Pitch cone angle - Gear: $\Gamma = 63.43$ degrees Outer cone distance: $A_o = 2.7951$ in Pitch Line Speed: $v_t = 818$ ft/min Transmitted Load: $W_t = 141$ lb		Factors in Design Analysis: Load distribution factor, K_m : Select from: Both gears straddle mounted: 1.00 One gear straddle mounted: 1.10 Neither gear straddle mounted: 1.25 Enter Factor, K_{mb} : 1.25 From Equation 10-16 Factor, K_{mb} : 1.25 Overload Factor: $K_o = 2.00$ Bending Size Factor: $K_s = 0.51$ Dynamic Factor: $K_v = 1.305$ Pitting Size Factor: $C_s = 0.53$ For $F < 0.50$, $C_s = 0.5$ Enter $C_s = 0.53$ Service Factor: $S_F = 1.00$ Bending Reliability Factor: $K_R = 1.00$ Pitting: $C_R = 1.00$ Enter: Design Life: 15000 hours Pinion - Number of load cycles: $N_P = 1.13E+09$ Gear - Number of load cycles: $N_G = 5.63E+08$ Bending Stress Cycle Factor: $K_L = 0.94$ Bending Stress Cycle Factor: $K_L = 0.95$ Pitting Stress Cycle Factor: $C_L = 1.27$ Pitting Stress Cycle Factor: $C_L = 1.36$													
Secondary Input Data: Face Width Guidelines (in): Enter: Face Width: $F = 0.839$ Enter: Elastic Coefficient: $C_p = 2300$ Enter: Quality Number: $A_v = 10$		Table 9-7 Figure 10-13 (for $P_d < 16$) Computed: Table 9-9 For $0.50 < F < 3.14$ For $F > 3.14$, $C_s = 0.83$ Use 1.00 if no unusual conditions Pitting: $C_R = 1.00$													
Face Width Guidelines (in): Enter: Face Width: $F = 0.839$ Enter: Elastic Coefficient: $C_p = 2300$ Enter: Quality Number: $A_v = 10$		For $R =$ <table border="1"> <tr> <td>K_R</td> <td>C_R</td> </tr> <tr> <td>0.9</td> <td>0.85</td> </tr> <tr> <td>0.99</td> <td>1.00</td> </tr> <tr> <td>0.999</td> <td>1.25</td> </tr> <tr> <td>0.9999</td> <td>1.50</td> </tr> <tr> <td>0.9999</td> <td>1.50</td> </tr> </table>		K_R	C_R	0.9	0.85	0.99	1.00	0.999	1.25	0.9999	1.50	0.9999	1.50
K_R	C_R														
0.9	0.85														
0.99	1.00														
0.999	1.25														
0.9999	1.50														
0.9999	1.50														
Enter: Bending Geometry Factors: Pinion: $J_P = 0.258$ Gear: $J_G = 0.220$ Enter: Pitting Geometry Factor: $I = 0.083$		Through Hardened Grade 1 Steel HB 266 HB 316 HB 198 HB 180													
Enter: Bending Geometry Factors: Pinion: $J_P = 0.258$ Gear: $J_G = 0.220$ Enter: Pitting Geometry Factor: $I = 0.083$		See Fig. 10-17 or Table 10-4 See Fig. 10-21 or Table 10-4 See Fig. 10-21 or Table 10-4													
Enter: Bending Geometry Factors: Pinion: $J_P = 0.258$ Gear: $J_G = 0.220$ Enter: Pitting Geometry Factor: $I = 0.083$		Stress Analysis - Bending: Pinion: Required $s_{at} = 13,808$ psi Gear: Required $s_{at} = 16,023$ psi Stress Analysis - Pitting: Pinion: Required $s_{ac} = 91,011$ psi Gear: Required $s_{ac} = 84,988$ psi Assumes $C_{xc} = 1.5$ for properly crowned teeth													
Enter: Bending Geometry Factors: Pinion: $J_P = 0.258$ Gear: $J_G = 0.220$ Enter: Pitting Geometry Factor: $I = 0.083$		Specify materials, alloy and heat treatment, for most severe requirement. One possible material specification: Pinion: HB 266 required: SAE 6150 OQT 1200; HB = 293 Gear: HB 316 required: SAE 6150 OQT 1100; HB = 341													

Problem 10-16**BEVEL GEARS****Forces and torque for shaft and bearing load analysis:****On Pinion Shaft - Torque:** $T_P = 370.59 \text{ lb-in}$ Mean radius of pinion: $r_m = 0.938 \text{ in}$ *Enter:* pressure angle: $\phi = 20 \text{ degrees}$ On Pinion - Tangential load: $W_{tP} = 395.2 \text{ lb}$ On Pinion - Radial load: $W_{rP} = 135.6 \text{ lb}$ On Pinion - Axial load: $W_{xP} = 47.9 \text{ lb}$ **On Gear Shaft - Torque:** $T_G = 1050.0 \text{ lb-in}$ On Gear - Tangential load: $W_{tG} = 395.2 \text{ lb}$ On Gear - Radial load: $W_{rG} = 47.9 \text{ lb}$ On Gear - Axial load: $W_{xG} = 135.6 \text{ lb}$ **See following page for stress analysis and design details.**

DESIGN OF BEVEL GEARS		APPLICATION:																					
Initial Input Data: Input Power: $P = 5$ hp Input Speed: $n_P = 850$ rpm Diametral Pitch: $P_d = 8$ Number of Pinion Teeth: $N_P = 18$ Desired Output Speed: $n_G = 300$ rpm Computed number of gear teeth: Enter: Chosen No. of Gear Teeth: $N_G = 51$		Conveyor with heavy shock driven by a gasoline engine Problem 10-16 Both gears straddle mounted																					
Computed data: Actual Output Speed: $n_G = 300.0$ rpm Gear Ratio: $m_G = 2.83$ Pitch Diameter - Pinion: $D_P = 2.250$ in Pitch Diameter - Gear: $D_G = 6.375$ in Pitch cone angle - Pinion: $\gamma = 19.44$ degrees Pitch cone angle - Gear: $\Gamma = 70.56$ degrees Outer cone distance: $A_o = 3.3802$ in Pitch Line Speed: $v_t = 501$ ft/min Transmitted Load: $W_t = 330$ lb		Factors In Design Analysis: Load distribution factor, K_m : Select from: Both gears straddle mounted: 1.00 One gear straddle mounted: 1.10 Neither gear straddle mounted: 1.25 Enter Factor, K_{mb} : 1.00 Factor, K_{mb} : 1.00 From Equation 10-16																					
Secondary Input Data: <table border="1"> <thead> <tr> <th></th> <th>Nom</th> <th>Max</th> <th>Max</th> </tr> </thead> <tbody> <tr> <td>Face Width Guidelines (in):</td> <td>1.014</td> <td>1.127</td> <td>1.250</td> </tr> <tr> <td>Enter: Face Width: $F =$</td> <td colspan="3">1.125 in</td> </tr> <tr> <td>Enter: Elastic Coefficient: $C_p =$</td> <td colspan="3">2300 Table 9-10</td> </tr> <tr> <td>Enter: Quality Number: $A_v =$</td> <td colspan="3">10 Table 9-3</td> </tr> </tbody> </table>			Nom	Max	Max	Face Width Guidelines (in):	1.014	1.127	1.250	Enter: Face Width: $F =$	1.125 in			Enter: Elastic Coefficient: $C_p =$	2300 Table 9-10			Enter: Quality Number: $A_v =$	10 Table 9-3			Overload Factor: $K_o = 2.00$ Table 9-7 Bending Size Factor: $K_s = 0.51$ Figure 10-13 (for $P_d < 16$) Dynamic Factor: $K_v = 1.241$ Computed: Table 9-9 Pitting Size Factor: $C_s = 0.58$ For $0.50 < F < 3.14$ For $F > 3.14$, $C_s = 0.83$ Service Factor: $S_F = 1.00$ Use 1.00 if no unusual conditions Bending Reliability Factor: $K_R = 1.00$ Pitting: $C_R = 1.00$	
	Nom	Max	Max																				
Face Width Guidelines (in):	1.014	1.127	1.250																				
Enter: Face Width: $F =$	1.125 in																						
Enter: Elastic Coefficient: $C_p =$	2300 Table 9-10																						
Enter: Quality Number: $A_v =$	10 Table 9-3																						
Face Width Guidelines (in): Enter: Face Width: $F =$ Enter: Elastic Coefficient: $C_p =$ Enter: Quality Number: $A_v =$		For K_v $B = 0.731$ $C = 65.0$																					
Face Width Guidelines (in): Enter: Face Width: $F =$ Enter: Elastic Coefficient: $C_p =$ Enter: Quality Number: $A_v =$		For $R =$ K_R C_R 0.9 0.85 0.62 0.99 1.00 1.00 0.999 1.25 1.12 0.9999 1.50 1.22																					
Stress Analysis - Bending: Pinion: Required $s_{at} = 13,300$ psi Gear: Required $s_{at} = 15,628$ psi Stress Analysis - Pitting: Assumes $C_{xc} = 1.5$ for properly crowned teeth Pinion: Required $s_{ac} = 133,166$ psi Gear: Required $s_{ac} = 125,768$ psi		Through Hardened Grade 1 Steel HB 255 Fig. 10-17 HB 307 Fig. 10-17 HB 321 Fig. 10-21 HB 300 Fig. 10-21																					
Specify materials, alloy and heat treatment, for most severe requirement. One possible material specification: Pinion: HB 321 required: SAE 6150 OQT 1100; HB = 341 Gear: HB 307 required: SAE 6150 OQT 1100; HB = 341																							

Problem 10-17**BEVEL GEARS****Forces and torque for shaft and bearing load analysis:****On Pinion Shaft - Torque:** $T_P = 26.25$ lb-inMean radius of pinion: $r_m = 0.386$ in*Enter:* pressure angle: $\phi = 20$ degreesOn Pinion - Tangential load: $W_{tP} = 68.0$ lbOn Pinion - Radial load: $W_{rP} = 23.9$ lbOn Pinion - Axial load: $W_{xP} = 6.3$ lb**On Gear Shaft - Torque:** $T_G = 99.2$ lb-inOn Gear - Tangential load: $W_{tG} = 68.0$ lbOn Gear - Radial load: $W_{rG} = 6.3$ lbOn Gear - Axial load: $W_{xG} = 23.9$ lb**See following page for stress analysis and design details.**

DESIGN OF BEVEL GEARS		APPLICATION:		Reciprocating saw driven by an electric motor	
Problem 10-17		Both gears straddle mounted			
Initial Input Data:		Factors in Design Analysis:			
Input Power: $P = 0.75$ hp		Load distribution factor, K_m :	From Equation 10-16		
Input Speed: $n_P = 1800$ rpm		Select from:	Factor, K_{mb}		
Diametral Pitch: $P_d = 20$		Both gears straddle mounted:	1.00		
Number of Pinion Teeth: $N_P = 18$		One gear straddle mounted:	1.10		
Desired Output Speed: $n_G = 475$ rpm		Neither gear straddle mounted:	1.25		
Computed number of gear teeth: 68.2		Enter Factor, K_{mb} :	1.00		
Enter: Chosen No. of Gear Teeth: $N_G = 68$		$K_m = 1.00$			
Computed data:		Table 9-7			
Actual Output Speed: $n_G = 476.5$ rpm		Overload Factor: $K_o = 1.75$	Figure 10-13 (for $P_d < 16$)		
Gear Ratio: $m_G = 3.78$		Bending Size Factor: $K_s = 0.50$	Computed: Table 9-9		
Pitch Diameter - Pinion: $D_P = 0.900$ in		Dynamic Factor: $K_v = 1.277$	For $0.50 < F < 3.14$		
Pitch Diameter - Gear: $D_G = 3.400$ in		Pitting Size Factor: $C_s = 0.5$	For $F > 3.14$, $C_s = 0.83$		
Pitch cone angle - Pinion: $\gamma = 14.83$ degrees		Enter $C_s = 0.5$			
Pitch cone angle - Gear: $\Gamma = 75.17$ degrees		Service Factor: $S_F = 1.00$	Use 1.00 if no unusual conditions		
Outer cone distance: $A_o = 1.7586$ in		Bending Reliability Factor: $K_R = 1.00$	Pitting: $C_R = 1.00$		
Pitch Line Speed: $v_t = 424$ ft/min		Enter: Design Life: 15000 hours	See Table 9-7		
Transmitted Load: $W_t = 58$ lb		Pinion - Number of load cycles: $N_P = 1.62E+09$	Y_N, Z_N : Pinion-Fig. 10-16; Gear-Fig. 10-20		
Secondary Input Data:		Gear - Number of load cycles: $N_G = 4.29E+08$	$N < 10^3$ $10^3 < N < 10^7$ $N > 10^7$		
Face Width Guidelines (in):	Nom	Bending Stress Cycle Factor: $K_L = 0.93$	2.70	0.49	0.93
	Max	Bending Stress Cycle Factor: $K_L = 0.95$	2.70	0.58	0.95
Enter: Face Width: $F = 0.500$ in		Pitting Stress Cycle Factor: $C_L = 0.97$	2.00	0.97	
Enter: Elastic Coefficient: $C_p = 2300$		Pitting Stress Cycle Factor: $C_L = 1.05$	2.00	1.05	
Enter: Quality Number: $A_v = 11$		Stress Analysis - Bending:			
Enter: Bending Geometry Factors:		Pinion: Required $s_{at} = 11,171$ psi	See Fig. 10-17 or Table 10-4		
		Gear: Required $s_{at} = 13,018$ psi			
Enter: Pitting Geometry Factor: $I = 0.085$		Stress Analysis - Pitting: Assumes $C_{xc} = 1.5$ for properly crowned teeth			
		Pinion: Required $s_{ac} = 119,964$ psi	See Fig. 10-21 or Table 10-4		
		Gear: Required $s_{ac} = 110,824$ psi			
Specify materials, alloy and heat treatment, for most severe requirement.					
One possible material specification:					
Pinion: HB 283 required: SAE 6150 OQT 1200; HB = 293					
Gear: HB 256 required: SAE 6150 OQT 1200; HB = 293					

WORMGEARING DATA FROM PROB. 8-52: $T_o = 924 \text{ LB}\cdot\text{IN.}$, $m_g = 30 \text{ RPM}$

FORCES: $W_{tG} = W_{xW} = T_o / (D_g/2) = 924 \text{ LB}\cdot\text{IN.} / 2.00 \text{ IN.} = 462 \text{ LB}$

PITCH LINE SPEED OF GEAR $= N_{tG} = \pi D_g m_g / 12 = \pi (4.00 \times 30) / 12 = 31.4 \text{ FT/MIN.}$

SLIDING VELOCITY $= N_s = N_{tG} / \sin \lambda = 31.4 / \sin(4.57) = 394 \text{ FT/MIN.}$

FROM FIG. 10-25: $\mu = 0.0323$ [COMPUTED FROM EQ. 10-27]

$W_{xG} = W_{tW} = 462 \times \frac{\cos(4.5) \sin(4.57) + 0.0323 \cos(4.57)}{\cos(4.5) \cos(4.57) - 0.0323 \sin(4.57)} = 53 \text{ LB}$
(EQ. 10-30)

$W_{rG} = W_{rW} = \frac{462 \sin(4.5)}{\cos(4.5) \cos(4.57) - 0.0323 \sin(4.57)} = 120 \text{ LB}$
(EQ. 10-31)

FRICTION FORCE $= W_f = \frac{(0.0323)(462)}{\cos(4.57) \cos(4.5) - 0.0323 \sin(4.57)} = 15.5 \text{ LB}$
(EQ. 10-32)

FRICTION POWER LOSS $= P_L = \frac{N_s W_f}{33000} = \frac{(394)(15.5)}{33000} = 0.185 \text{ HP}$

INPUT POWER $= P_{iH} = P_o + P_L = \frac{T_o m_g}{63000} + 0.185 = 0.44 + 0.185 = 0.625 \text{ HP}$

EFFICIENCY $= P_o / P_{iH} \times 100\% = 70.4\%$; INPUT SPEED $= m_g \cdot V_R = (30)(40) = 1200 \text{ RPM}$

STRESS $\sigma_G = \frac{W_d}{A_F F_m} = \frac{W_{tG}}{K_s A_F F \pi \cos \lambda} = \frac{(462)(10)}{(0.974)(0.100)(0.625)(\pi)(\cos 4.57)} = 24235 \text{ PSI}$

$K_v = 1200 / (1200 + N_{tG}) = 1200 / (1200 + 31.4) = 0.974$

PITTING $W_{tR} = C_s D_g^{0.8} F_e C_m C_v = (1000)(4.00)^{0.8} (0.625)(0.814)(0.974) = 659 \text{ LB}$
(FIG. 12-21)

BECAUSE $W_{tR} > W_{tG}$ - OK FOR PITTING.

$\sigma_G = 24235 \text{ PSI}$ SLIGHTLY HIGHER THAN $S_{at} = 24000 \text{ PSI}$ FOR PHOSPHOR BRONZE

(SEE SPREADSHEET SOLUTION ON FOLLOWING PAGE.)

Wormgearing - Design		Problem: 10-18	
Input Data:			
Desired output torque:	$T_o = 924$ lb-in		
Output speed:	$n_g = 30$ rpm		
Velocity Ratio:	$VR = 40$		
Design Decisions:			
Diametral pitch:	$P_d = 10$		
No. of worm threads:	$N_w = 1$		
Required No. of gear teeth:	$N_g = 40$		
Specify No. of gear teeth:	$N_g = 40$		
Normal pressure angle:	$\phi_n = 14.5$ degrees		
Computed Results and Additional Inputs:			
Actual input speed:	$n_w = 1200$ rpm		
Actual velocity ratio:	$VR = 40$		
Gear pitch diameter:	$D_g = 4$ in		
Specify worm diameter:	$D_w = 1.25$ in		
Actual center distance:	$C = 2.625$ in		
	$C^{0.875}/D_w = 1.86$		
	Should be >1.6 and <3.0		
Circular pitch of gear:	$p_g = 0.314$ in		
Axial pitch of worm:	$p_{xw} = 0.314$ in		
Lead of the worm:	$L = 0.314$ in		
Lead angle:	$\lambda = 4.574$ deg		
Addendum:	$a = 0.100$ in		
Dedendum:	$b = 0.116$ in		
Worm outside diameter:	$D_{ow} = 1.450$ in		
Worm root diameter:	$D_{rw} = 1.019$ in		
Nominal worm face length:	$F_{whom} = 1.789$ in		
Gear throat diameter:	$D_{tg} = 4.200$ in		
Nominal gear face width:	$F_{eg} = 0.735$ in		
Max effective gear face width:	$0.67 \cdot D_w = 0.8375$ in		
Effective gear face width:	$F_e = 0.625$ in		
	(Used given face width)		
Notes: 1. Bending stress on gear slightly high 2. Suggest using larger face width; Say $F = F_e = 0.75$ in.			
		Additional Computed Results:	
		Pitch line speed - Gear: 31.42 ft/min	
		Sliding velocity $v_s = 394$ ft/min	
		Coefficient of friction: 0.032 if $v_s > 10$ ft/min	
		Forces: (lb)	
		Gear	Worm
		Tangential: 462	53
		Radial: 120	120
		Axial: 53	462
		Friction force, $W_f = 15.6$ lb	
		Power:	
		Power output from gear:	0.440 hp
		Power loss - friction:	0.186 hp
		Power Input:	0.626 hp
		Efficiency:	70.3 %
		Stresses:	
		Bending Stress on Gear:	
		Enter: Lewis form factor: $y = 0.100$	----->
		Normal circular pitch: 0.313 in	
		Dynamic factor: $K_v = 0.974$	
		Bending stress on gear: 24223 psi [Using effective gear face width]	
		Allowable stresses-Bronze: Manganese = 17000 psi; Phosphor = 24000 psi	
		Surface Durability: [Hardened steel worm; bronze gear]	
		Type of bronze: $D_g \rightarrow$	<2.5 in >2.5 in >8 in >25 in <25 in
		Sand cast: $C_s = 903$	1000
		Chill cast or forged: $C_s =$	1137 1000
		Centrifugally cast: $C_s =$	1143 1000
		Enter: Materials factor: $C_s = 1000$	
		Gear Ratio: $m_g = 6$ to 20	20 to 76 >76 Actual $m_g = 40$
		Ratio correction factor: $C_m = \#NUM!$	0.814 0.885
		Enter: $C_m = 0.814$	
		Sliding velocity: <700	700-3000 >3000 Actual $v_s = 394$
		Velocity factor: $C_v = 0.427$	0.439 0.642
		Enter: $C_v = 0.427$	
		Rated tangential load: $W_{tr} = 659$ lb	
		Must be $> W_t = 462$ lb	
		3. Equation for C_m produces invalid result for large gear ratio as $VR = 40$	

Wormgearing - Design		Problem: 10-18A Adjusted $F_o = 0.750$ in	
Input Data:			
Desired output torque:		$T_o =$	924 lb-in
Output speed:		$n_g =$	30 rpm
Velocity Ratio:		$VR =$	40
Design Decisions:			
Diametral pitch:		$P_d =$	10
No. of worm threads:		$N_w =$	1
Required No. of gear teeth:		$N_g =$	40
Specify No. of gear teeth:		$N_g =$	40
Normal pressure angle:		$\phi_n =$	14.5 degrees
Computed Results and Additional Inputs:			
Actual input speed:		$n_w =$	1200 rpm
Actual velocity ratio:		$VR =$	40
Gear pitch diameter:		$D_g =$	4 in
Specify worm diameter:		$D_w =$	1.25 in
Actual center distance:		$C =$	2.625 in
		$C^{0.875}/D_w =$	1.86
Should be >1.6 and <3.0			
Circular pitch of gear:		$P_g =$	0.314 in
Axial pitch of worm:		$P_{xw} =$	0.314 in
Lead of the worm:		$L =$	0.314 in
Lead angle:		$\lambda =$	4.574 deg
Addendum:		$a =$	0.100 in
Dedendum:		$b =$	0.116 in
Worm outside diameter:		$D_{ow} =$	1.450 in
Worm root diameter:		$D_{rw} =$	1.019 in
Nominal worm face length:		$F_{whom} =$	1.789 in
Gear throat diameter:		$D_{tg} =$	4.200 in
Nominal gear face width:		$F_{eg} =$	0.735 in
Max effective gear face width:		$0.67 \cdot D_w =$	0.8375 in
Effective gear face width:		$F_o =$	0.750 in
(Used given face width)			
Notes: 1. Bending stress on gear OK for Phosphor Bronze 2. Using $F = F_o = 0.75$ in.			
Additional Computed Results:			
Pitch line speed - Gear:		31.42	ft/min
Sliding velocity $v_s =$		394	ft/min
Coefficient of friction:		0.032	If $v_s > 10$ ft/min
Forces: (lb)		Gear	Worm
Tangential:		462	53
Radial:		120	120
Axial:		53	462
Friction force, $W_f =$		15.6	lb
Power:			
Power output from gear:		0.440	hp
Power loss - friction:		0.186	hp
Power input:		0.626	hp
Efficiency:		70.3	%
Stresses:			
Bending Stress on Gear:			
Enter: Lewis form factor: $y =$		0.100	----->
Normal circular pitch:		0.313	in
Dynamic factor: $K_v =$		0.974	
Bending stress on gear:		20186	psi [Using effective gear face width]
Allowable stresses-Bronze: Manganese = 17000 psi, Phosphor = 24000 psi			
Surface Durability: [Hardened steel worm; bronze gear]			
Type of bronze/ D_g --->		>2.5 in	<2.5 in
Sand cast: $C_s =$		903	1000
Chill cast or forged: $C_s =$			
Centrifugally cast: $C_s =$			
Enter: Materials factor: $C_s =$		1000	
Gear Ratio: $m_g =$		6 to 20	20 to 76
Ratio correction factor: $C_m =$		#NUM!	0.814
Enter: $C_m =$		0.814	
Sliding velocity:		<700	$700-3000$
Velocity factor: $C_v =$		0.427	0.439
Enter: $C_v =$		0.427	
Rated tangential load: $W_R =$		790	lb
Must be $> W_f =$		462	lb
3. Equation for C_m produces invalid result for large gear ratio as $VR = 40$			

Problems 10-19 and 10-20

COMPARISON OF THREE PROPOSED DESIGNS

See details on following three spreadsheets

Given data:

Diametral pitch, $P_d =$	12	
Velocity ratio, $VR =$	20	
Output speed (Gear) =	90 rpm	
Worm pitch diameter, $D_w =$	1.000 in	See comment
Gear face width, $F =$	0.500 in	See comment
Normal pressure angle, $\phi_n =$	14.5 degrees	

Assumed gear is made from chilled cast phosphor bronze

Allowable bending stress = 24,000 psi

Results:**DESIGN**

	A	B	C	
Number of threads in worm:	1	2	4	
Output torque (lb-in), $T_o =$	202	484	878	
Output power (hp), $P_o =$	0.289	0.691	1.254	
Gear bending stress (psi)	19190	23963	23987	Limits in Bold
Allowable bending stress (psi)	24000	24000	24000	
Rated load for surface durability (lb)	242	418	714	Limits in Bold
Gear transmitted load (lb)	242	290	263	
Efficiency (%) [Problem 20]	72.9	84.1	90.8	
Power input (hp)	0.396	0.822	1.381	
Lead angle (degrees)	4.76	9.46	18.4	
Gear pitch diameter (in)	1.667	3.333	6.667	
Center distance (in)	1.333	2.167	3.833	

Comments on results:The given face width is small. Could use $F > 0.601$ in to maximize effective face width.

Worm diameter is too large for Design A. See Equations 10-46 and 10-47

Worm diameter is too small for Design C. See Equations 10-46 and 10-47

Design A is limited by surface durability

Designs B and C are limited by bending stress in gear teeth.

As number of threads in worm increases:

Lead angle increases

Efficiency increases

Torque and power capacity increase

BUT: Gear size and center distance increase

Wormgearing - Design		Problem: 16-15a and 20a	
Input Data:			
Desired output torque:		$T_o = 202 \text{ lb-in}$	
Output speed:		$n_o = 90 \text{ rpm}$	
Velocity Ratio:		$VR = 20$	
Design Decisions:			
Diametral pitch:		$P_d = 12$	
No. of worm threads:		$N_w = 1$	
Required No. of gear teeth:		$N_g = 20$	
Specify No. of gear teeth:		$N_g = 20$	
Normal pressure angle:		$\phi_n = 14.5 \text{ degrees}$	
Computed Results and Additional Inputs:			
Actual input speed:		$n_w = 1800 \text{ rpm}$	
Actual velocity ratio:		$VR = 20$	
Gear pitch diameter:		$D_g = 1.6667 \text{ in}$	
Specify worm diameter:		$D_w = 1.000 \text{ in}$	
Actual center distance:		$C = 1.333 \text{ in}$	
<p>Use smaller worm diameter Should be >1.6 and <3.0</p> <p>$C^{0.875}/D_w = 1.29$ LOW</p>			
Circular pitch of gear:		$p_g = 0.262 \text{ in}$	
Axial pitch of worm:		$p_{xw} = 0.262 \text{ in}$	
Lead of the worm:		$L = 0.262 \text{ in}$	
Lead angle:		$\lambda = 4.764 \text{ deg}$	
Addendum:		$a = 0.083 \text{ in}$	
Dedendum:		$b = 0.096 \text{ in}$	
Worm outside diameter:		$D_{ow} = 1.167 \text{ in}$	
Worm root diameter:		$D_{rw} = 0.807 \text{ in}$	
Nominal worm face length:		$F_{whom} = 1.054 \text{ in}$	
Gear throat diameter:		$D_{tg} = 1.833 \text{ in}$	
Nominal gear face width:		$F_{es} = 0.601 \text{ in}$	
Max effective gear face width:		$0.67 \cdot D_w = 0.670 \text{ in}$	
Effective gear face width:		$F_e = 0.500 \text{ in}$	
<p>Given face width is small; Could use $F > 0.601$</p> <p>[Used given face width]</p>			
Additional Computed Results:			
Pitch line speed - Gear:		39.27 ft/min	
Sliding velocity $v_s =$		473 ft/min	
Coefficient of friction:		0.030 If $v_s > 10 \text{ ft/min}$	
Forces: (lb)			
	Gear	Worm	
Tangential:	242	28	
Radial:	63	63	
Axial:	28	242	
Friction force, $W_f =$	7.5 lb		
Power:			
Power output from gear:	0.289 hp		
Power loss - friction:	0.107 hp		
Power input:	0.396 hp		
Efficiency:	72.9 %		
Stresses:			
Bending Stress on Gear:			
Enter: Lewis form factor $y =$	$0.100 \rightarrow$		
Normal circular pitch:	0.261 in		
Dynamic factor: $K_v =$	0.988		
Bending stress on gear: 19190 psi [Using effective gear face width]			
Allowable stresses-Bronze: Manganese = 17000 psi; Phosphor = 24000 psi			
Surface Durability: [Hardened steel worm; bronze gear]			
Type of bronze/ $D_g \rightarrow$	$>2.5 \text{ in}$	$<2.5 \text{ in}$	$>8 \text{ in}$
Sand cast: $C_s =$	1084	1000	
Chill cast or forged: $C_s =$			
Centrifugally cast: $C_s =$		1311	1000
Enter: Materials factor $C_s =$	1000		
Gear Ratio: $m_g =$	6 to 20	20 to 76	>76 Actual $m_g = 20$
Ratio correction factor: $C_m =$	0.820	0.819	1.017
Enter: $C_m =$	0.819		
Sliding velocity:	<700	700-3000	>3000 Actual $v_s = 473$
Velocity factor: $C_v =$	0.392	0.395	0.557
Enter: $C_v =$	0.392		
Rated tangential load: $W_R =$	242 lb		
Must be $> W_f =$	242 lb		
Adjusted output torque until limits reached on either bending or surface durability			
Surface durability controls this design			

Wormgearing - Design		Problem: 10-195 and 20b	
Input Data:			
Desired output torque:	$T_c =$	484 lb-in	
Output speed:	$n_g =$	90 rpm	
Velocity Ratio:	$VR =$	20	
Design Decisions:			
Diametral pitch:	$P_d =$	12	
No. of worm threads:	$N_w =$	2	
Required No. of gear teeth:	$N_g =$	40	
Specify No. of gear teeth:	$N_g =$	40	
Normal pressure angle:	$\phi_p =$	14.5 degrees	
Computed Results and Additional Inputs:			
Actual input speed:	$n_w =$	1800 rpm	
Actual velocity ratio:	$VR =$	20	
Gear pitch diameter:	$D_g =$	3.33333 in	
Specify worm diameter:	$D_w =$	1.930 in	
Actual center distance:	$C =$	2.167 in	
	$C_{0.975}/D_w =$	1.97	
	Should be >1.6 and <3.0		
Circular pitch of gear:	$p_g =$	0.262 in	
Axial pitch of worm:	$p_{xw} =$	0.262 in	
Lead of the worm:	$L =$	0.524 in	
Lead angle:	$\lambda =$	9.462 deg	
Addendum:	$a =$	0.083 in	
Dedendum:	$b =$	0.096 in	
Worm outside diameter:	$D_{ow} =$	1.167 in	
Worm root diameter:	$D_{rw} =$	0.807 in	
Nominal worm face length:	$F_{whom} =$	1.491 in	
Gear throat diameter:	$D_{tg} =$	3.500 in	
Nominal gear face width:	$F_{ag} =$	0.601 in	
Max effective gear face width:	$0.67 \cdot D_w =$	0.670 in	
Effective gear face width:	$F_g =$	0.500 in	
[Used given face width]			
Given face width is small; Could use $F >$ 0.601 in			
Additional Computed Results:			
Pitch line speed - Gear:	78.54	ft/min	
Sliding velocity $v_s =$	478	ft/min	
Coefficient of friction:	0.030	If $v_s > 10$ ft/min	
Forces: (lb)			
	Gear	Worm	
Tangential:	290	58	
Radial:	77	77	
Axial:	58	290	
Friction force, $W_f =$	9.1	lb	
Power:			
Power output from gear:	0.691	hp	
Power loss - friction:	0.131	hp	
Power Input:	0.822	hp	
Efficiency: 84.1 %			
Stresses:			
Bending Stress on Gear:			
Enter: Lewis form factor $y =$	0.103	→	
Normal circular pitch:	0.258	in	
Dynamic factor: $K_v =$	0.939		
Bending stress on gear:	23963	psi [Using effective gear face width]	
Allowable stresses-Bronze: Manganese = 17000 psi; Phosphor = 24000 psi			
Surface Durability: [Hardened steel worm; bronze gear]			
Type of bronze/ $D_g \rightarrow$	>2.5 in	<2.5 in	>8 in
Sand cast: $C_s =$	940	1000	
Chill cast or forged: $C_s =$			
Centrifugally cast: $C_s =$			
Enter: Materials factor: $C_m =$	1000	Chilled Cast - Phosphor bronze	
Gear Ratio: $m_g =$	6 to 20	20 to 76	Actual $m_g = 20$
Ratio correction factor: $C_m =$	0.820	0.819	1.017
Enter: $C_{m1} =$	0.819		
Sliding velocity:	<700	700-3000	>3000
Velocity factor: $C_v =$	0.390	0.393	0.553
Enter: $C_v =$	0.390		
Rated tangential load: $W_R =$	418	lb	
Must be $> W_f =$	290	lb	
Adjusted output torque until limits reached on either bending or surface durability			
Bending stress controls this design			

Wormgearing - Design		Problem: (8-19c and 20c)	
Input Data:			
Desired output torque:	$T_d =$	678 lb-in	
Output speed:	$n_g =$	90 rpm	
Velocity Ratio:	$VR =$	20	
Design Decisions:			
Diametral pitch:	$P_d =$	12	
No. of worm threads:	$N_w =$	4	
Required No. of gear teeth:	$N_g =$	80	
Specify No. of gear teeth:	$N_g =$	80	
Normal pressure angle	$\phi_n =$	14.5 degrees	
Computed Results and Additional Inputs:			
Actual input speed:	$n_w =$	1800 rpm	
Actual velocity ratio:	$VR =$	20	
Gear pitch diameter:	$D_g =$	6.6667 in	
Specify worm diameter:	$D_w =$	1.000 in	
Actual center distance:	$C =$	3.833 in	
Use larger worm diameter Should be >1.6 and <3.0			
Circular pitch of gear:	$p_g =$	0.262 in	
Axial pitch of worm:	$p_w =$	0.262 in	
Lead of the worm:	$L =$	1.047 in	
Lead angle:	$\lambda =$	18.435 deg	
Addendum:	$a =$	0.083 in	
Dedendum:	$b =$	0.096 in	
Worm outside diameter:	$D_{ow} =$	1.167 in	
Worm root diameter:	$D_{rw} =$	0.807 in	
Nominal worm face length:	$F_{worm} =$	2.108 in	
Gear throat diameter:	$D_{tg} =$	6.833 in	
Nominal gear face width:	$F_{eg} =$	0.601 in	
Max effective gear face width:	$0.67 \cdot D_w =$	0.670 in	
Effective gear face width:	$F_e =$	0.500 in	
[Used given face width]			
Given face width is small; Could use $F > 0.601$ in			
Additional Computed Results:			
Pitch line speed - Gear:	157.08	ft/min	
Sliding velocity $v_s =$	497	ft/min	
Coefficient of friction:	0.029	If $v_s > 10$ ft/min	
Forces: (lb)			
	Gear	Worm	
Tangential:	263	97	
Radial:	73	73	
Axial:	97	263	
Friction force, $W_f =$	8.4	lb	
Power:			
Power output from gear:	1.254	hp	
Power loss - friction:	0.127	hp	
Power input:	1.381	hp	
Efficiency: 90.8 %			
Stresses:			
Bending Stress on Gear:			
Enter: Lewis form factor $y =$	0.403	→	
Normal circular pitch:	0.248	in	
Dynamic factor: $K_v =$	0.894		
Bending stress on gear:	23987	psi [Using effective gear face width]	
Allowable stresses-Bronze: Manganese = 17000 psi; Phosphor = 24000 psi			
Surface Durability: [Hardened steel worm; bronze gear]			
Type of bronze/ $D_g \rightarrow$	>2.5 in	<2.5 in	>8 in <8 in >25 in <25 in
Sand cast: $C_s =$	797	1000	
Chill cast or forged: $C_s =$			1036 1000
Centrifugally cast: $C_s =$			1103 1000
Enter: Materials factor $C_p =$	1000	Chilled Cast - Phosphor bronze	
Gear Ratio: $m_g =$	6 to 20	20 to 76	Actual $m_g = 20$
Ratio correction factor: $C_m =$	0.820	0.819	1.017
Enter: $C_m = 0.819$			
Sliding velocity:	<700	700-3000	>3000 Actual $v_s = 497$
Velocity factor: $C_v =$	0.382	0.384	0.537
Enter: $C_v = 0.382$			
Rated tangential load: $W_R =$	714	lb	
Must be $> W_f =$	263	lb	
Adjusted output torque until limits reached on either bending or surface durability			
Bending stress controls this design			

Wormgearing - Design		Problem: 10-21	
Input Data:			
Desired output torque:	$T_o = 984 \text{ lb-in}$		
Output speed:	$n_g = 80 \text{ rpm}$		
Velocity Ratio:	$VR = 7.5$		
Design Decisions:			
Diametral pitch:	$P_d = 8$		
No. of worm threads:	$N_w = 4$		
Required No. of gear teeth:	$N_g = 30$		
Specify No. of gear teeth:	$N_g = 30$		
Normal pressure angle:	$\phi_n = 14.5 \text{ degrees}$		
Computed Results and Additional Inputs:			
Actual input speed:	$n_w = 600 \text{ rpm}$		
Actual velocity ratio:	$VR = 7.5$		
Gear pitch diameter:	$D_g = 3.75 \text{ in}$		
Specify worm diameter:	$D_w = 1.375 \text{ in}$		
Actual center distance:	$C = 2.563 \text{ in}$		
	$C^{0.875}/D_w = 1.66$		
	Should be >1.6 and <3.0		
Circular pitch of gear:	$P_g = 0.393 \text{ in}$		
Axial pitch of worm:	$P_{xw} = 0.393 \text{ in}$		
Lead of the worm:	$L = 1.571 \text{ in}$		
Lead angle:	$\lambda = 19.983 \text{ deg}$		
Addendum:	$a = 0.125 \text{ in}$		
Dedendum:	$b = 0.145 \text{ in}$		
Worm outside diameter:	$D_{ow} = 1.625 \text{ in}$		
Worm root diameter:	$D_{rw} = 1.086 \text{ in}$		
Nominal worm face length:	$F_{wnom} = 1.936 \text{ in}$		
Gear throat diameter:	$D_{tg} = 4.000 \text{ in}$		
Nominal gear face width:	$F_{eg} = 0.866 \text{ in}$		
Max effective gear face width:	$0.67 \cdot D_w = 0.92125 \text{ in}$		
Effective gear face width:	$F_e = 0.866 \text{ in}$		
Additional Computed Results:			
Pitch line speed - Gear:	78.54 ft/min		
Sliding velocity $v_s =$	230 ft/min		
Coefficient of friction:	$0.041 \text{ if } v_s > 10 \text{ ft/min}$		
Forces: (lb)			
	Gear	Worm	
Tangential:	525	216	
Radial:	147	147	
Axial:	216	525	
Friction force, $W_f =$	24.0 lb		
Power:			
Power output from gear:	1.250 hp		
Power loss - friction:	0.167 hp		
Power input:	1.416 hp		
Efficiency:	88.2 %		
Stresses:			
Bending Stress on Gear:			
Enter Lewis form factor, y	$y = 0.100$		
Normal circular pitch:	0.369 in		
Dynamic factor: $K_v =$	0.939		
Bending stress on gear: 17495 psi [Using effective gear face width]			
Allowable stresses-Bronze: Manganese = 17000 psi; Phosphor = 24000 psi			
Surface Durability: [Hardened steel worm; bronze gear]			
Type of bronze/ $D_g \rightarrow$	$>2.5 \text{ in}$	$<2.5 \text{ in}$	$>8 \text{ in}$
	$<8 \text{ in}$	$>25 \text{ in}$	$<25 \text{ in}$
Sand cast: $C_s =$	916	1000	
Chill cast or forged: $C_s =$			1150 1000
Centrifugally cast: $C_s =$			1148 1000
Enter Materials factor $C_m = 1000$			
Gear Ratio: $m_g =$	6 to 20	20 to 76	>76
			Actual $m_g = 7.5$
Ratio correction factor: $C_m =$	0.719	0.794	1.099
Enter $C_m = 0.719$			
Sliding velocity:	<700	700-3000	>3000
			Actual $v_s = 230$
Velocity factor: $C_v =$	0.512	0.597	0.974
Enter $C_v = 0.512$			
Rated tangential load: $W_{tr} = 918 \text{ lb}$			
Must be $> W_f = 525 \text{ lb}$			

Wormgearing - Design		Problem: 10-22	
Input Data:			
Desired output torque:	$T_p =$	52.5 lb-in	
Output speed:	$n_g =$	600 rpm	
Velocity Ratio:	$VR =$	3	
Design Decisions:			
Diametral pitch:	$P_d =$	12	
No. of worm threads:	$N_w =$	8	
Required No. of gear teeth:	$N_g =$	18	
Specify No. of gear teeth:	$N_g =$	18	
Normal pressure angle:	$\phi_n =$	25 degrees	
Computed Results and Additional Inputs:			
Actual input speed:	$n_w =$	1800 rpm	
Actual velocity ratio:	$VR =$	3	
Gear pitch diameter:	$D_g =$	1.5 in	
Specify worm diameter:	$D_w =$	0.5 in	
Actual center distance:	$C =$	1.000 in	
	$C^{0.875}/D_w =$	2.00	
	Should be >1.6 and <3.0		
Circular pitch of gear:	$p_g =$	0.262 in	
Axial pitch of worm:	$p_{xw} =$	0.262 in	
Lead of the worm:	$L =$	1.571 in	
Lead angle:	$\lambda =$	45.000 deg	
Addendum:	$a =$	0.083 in	
Dedendum:	$b =$	0.096 in	
Worm outside diameter:	$D_{ow} =$	0.667 in	
Worm root diameter:	$D_{rw} =$	0.307 in	
Nominal worm face length:	$F_{wnom} =$	1.000 in	
Gear throat diameter:	$D_{ts} =$	1.667 in	
Nominal gear face width:	$F_{eg} =$	0.441 in	
Max effective gear face width:	$0.67 \cdot D_w =$	0.335 in	
Effective gear face width:	$F_e =$	0.335 in	
Additional Computed Results:			
Pitch line speed - Gear:	235.62 ft/min		
Sliding velocity $v_s =$	333 ft/min		
Coefficient of friction:	0.035 If $v_s > 10$ ft/min		
Forces: (lb)			
	Gear	Worm	
Tangential:	70	76	
Radial:	48	48	
Axial:	76	70	
Friction force, $W_f =$	4.0 lb		
Power:			
Power output from gear:	0.500 hp		
Power loss - friction:	0.040 hp		
Power input:	0.540 hp		
Efficiency: 92.6 %			
Stresses:			
Bending Stress on Gear:			
Enter Lewis form factor $y =$	0.59		
Normal circular pitch:	0.185 in		
Dynamic factor: $K_v =$	0.836		
Bending stress on gear:	9003 psi [Using effective gear face width]		
Allowable stresses-Bronze: Manganese = 17000 psi; Phosphor = 24000 psi			
Surface Durability: [Hardened steel worm; bronze gear]			
Type of bronze/ $D_g \rightarrow$	>2.5 in	<2.5 in	>8 in
Sand cast: $C_s =$	1106	1000	
Chill cast or forged: $C_s =$			1331
Centrifugally cast: $C_s =$			1000
Enter Materials factor $C_m =$	1000		
Gear Ratio: $m_g =$	6 to 20	20 to 76	>76
Ratio correction factor: $C_m =$	0.578	0.779	1.129
Enter $C_m =$	0.578		
Sliding velocity:	<700	700-3000	>3000
Velocity factor: $C_v =$	0.457	0.483	0.731
Enter $C_v =$	0.457		
Rated tangential load: $W_{tr} =$	122 lb		
Must be $> W_f =$	70 lb		

Wormgearing - Design		Problem: 18-23	
Input Data:			
Desired output torque:	$T_o =$	4200 lb-in	
Output speed:	$n_o =$	45 rpm	
Velocity Ratio:	$VR =$	10	
Design Decisions:			
Diametral pitch:	$P_d =$	8	
No. of worm threads:	$N_w =$	2	
Required No. of gear teeth:	$N_g =$	80	
Specify No. of gear teeth:	$N_g =$	89	
Normal pressure angle:	$\phi_n =$	14.5 degrees	
Computed Results and Additional Inputs:			
Actual input speed:	$n_w =$	1800 rpm	
Actual velocity ratio:	$VR =$	40	
Gear pitch diameter:	$D_g =$	10 in	
Specify worm diameter:	$D_w =$	2.25 in	
Actual center distance:	$C =$	6.125 in	
	$C^{0.875}/D_w =$	2.17	
	Should be >1.6 and <3.0		
Circular pitch of gear:	$p_g =$	0.393 in	
Axial pitch of worm:	$p_{xw} =$	0.393 in	
Lead of the worm:	$L =$	0.785 in	
Lead angle:	$\lambda =$	6.340 deg	
Addendum:	$a =$	0.125 in	
Dedendum:	$b =$	0.145 in	
Worm outside diameter:	$D_{ow} =$	2.500 in	
Worm root diameter:	$D_{rw} =$	1.961 in	
Nominal worm face length:	$F_{wnom} =$	3.162 in	
Gear throat diameter:	$D_{ts} =$	10.250 in	
Nominal gear face width:	$F_{ag} =$	1.090 in	
Max effective gear face width:	$0.67 \cdot D_w =$	1.5075 in	
Effective gear face width:	$F_e =$	1.090 in	
Additional Computed Results:			
Pitch line speed - Gear:	117.81	ft/min	
Sliding velocity $v_s =$	1067	ft/min	
Coefficient of friction:	0.020	If $v_s > 10$ ft/min	
Forces: (lb)			
	Gear	Worm	
Tangential:	840	111	
Radial:	219	219	
Axial:	111	840	
Friction force, $W_f =$	17.6	lb	
Power:			
Power output from gear:	3.000	hp	
Power loss - friction:	0.570	hp	
Power input:	3.570	hp	
Efficiency:	84.0	%	
Stresses:			
Bending Stress on Gear:			
Enter Lewis form factor $y =$	0.109		
Normal circular pitch:	0.390	in	
Dynamic factor: $K_v =$	0.911		
Bending stress on gear:	21689	psi [Using effective gear face width]	
Allowable stresses-Bronze: Manganese = 17000 psi; Phosphor = 24000 psi			
Surface Durability: [Hardened steel worm; bronze gear]			
Type of bronze/ $D_g \rightarrow$	>2.5 in	<2.5 in	>8 in
Sand cast: $C_s =$	713	1000	
Chill cast or forged: $C_s =$			956
Centrifugally cast: $C_s =$			1000
Enter Materials factor $C_m =$	$1/3$	Sand Cast	
Gear Ratio: $m_g =$	6 to 20	20 to 76	Actual $m_g = 40$
Ratio correction factor: $C_m =$	#NUM!	0.814	0.885
Enter $C_p =$	0.875		
Sliding velocity:	<700	$700-3000$	Actual $v_s = 1067$
Velocity factor: $C_v =$	0.204	0.248	0.297
Enter $C_v =$	0.248		
Rated tangential load: $W_R =$	990	lb	
Must be $> W_f =$	840	lb	

PROBLEM 10-24**COMPARISON OF DESIGNS A and B.**

**Note: Also includes a revised Design A with
a smaller worm diameter and larger gear face width**

Given data:	Design		
	A	B	A Revised
Diametral pitch:	6	10	6
Threads in worm:	1	2	1
Teeth in gear:	30	60	30
Worm diameter (in):	2.000	1.250	1.750 Design A Rev. - Smaller diameter
Face width - Gear (in):	1.000	0.625	1.130 Design A Rev. - Larger face width
Pressure angle (deg):	14.5	14.5	14.5
Results:			
Forces - Gear (lb):			
Tangential:	480	400	480
Radial:	125	106	125
Axial:	58	82	65
Forces - Worm (lb):			
Tangential:	58	82	65
Radial:	125	106	125
Axial:	480	400	480
Lead angle (degrees):	4.76	9.09	5.44 Design B - Higher lead angle
Efficiency (%):	69.1	77.6	70.6 Design B - Higher efficiency
Power output (hp):	0.381	0.381	0.381
Power input (hp):	0.552	0.491	0.54 Design B OK for 0.50 hp motor
Gear pitch diameter (in):	5.000	6.000	5.000 Design A smaller
Center distance (in):	3.500	3.625	3.375 Design A smaller
Stress - Wormgear (psi):	9400	21171	8324 Design A - Lower bending stress
Rated load - Durability (lb):	1193	937	1406
Design A - OK for sand cast Manganese Bronze			
Design B - OK for sand cast Phosphor bronze			

Wormgearing - Design		Problem: 16-24A	
Input Data:			
Desired output torque:	$T_g = 1200$ lb-in		
Output speed:	$n_g = 20$ rpm		
Velocity Ratio	$VR = 30$		
Design Decisions:			
Diametral pitch:	$P_d = 6$		
No. of worm threads:	$N_w = 1$		
Required No. of gear teeth:	$N_g = 30$		
Specify No. of gear teeth:	$N_g = 30$		
Normal pressure angle:	$\phi_n = 14.5$ degrees		
Computed Results and Additional Inputs:			
Actual input speed:	$n_w = 600$ rpm		
Actual velocity ratio:	$VR = 30$		
Gear pitch diameter:	$D_g = 5$ in		
Specify worm diameter:	$D_w = 2.000$ in		
Actual center distance:	$C = 3.500$ in		
$C^{0.875}/D_w = 1.50$ LOW			
Use smaller worm diameter Should be >1.6 and <3.0			
Circular pitch of gear:	$p_g = 0.524$ in		
Axial pitch of worm:	$p_{xw} = 0.524$ in		
Lead of the worm:	$L = 0.524$ in		
Lead angle:	$\lambda = 4.764$ deg		
Addendum:	$a = 0.167$ in		
Dedendum:	$b = 0.193$ in		
Worm outside diameter:	$D_{ow} = 2.333$ in		
Worm root diameter:	$D_{rw} = 1.614$ in		
Nominal worm face length:	$F_{wnom} = 2.582$ in		
Gear throat diameter:	$D_{ts} = 5.333$ in		
Nominal gear face width:	$F_{og} = 1.202$ in		
Max effective gear face width:	$0.67^*D_w = 1.340$ in		
Effective gear face width:	$F_g = 1.000$ in		
[Used given face width]			
Given face width is small; Could use $F > 1.202$			
Additional Computed Results:			
Pitch line speed - Gear:	26.18 ft/min		
Sliding velocity $v_s =$	315 ft/min		
Coefficient of friction:	0.036 If $v_s > 10$ ft/min		
Forces: (lb)			
	Gear	Worm	
Tangential:	480	58	
Radial:	125	125	
Axial:	58	480	
Friction force, $W_f =$	17.9 lb		
Power:			
Power output from gear:	0.381 hp		
Power loss - friction:	0.171 hp		
Power input:	0.552 hp		
Efficiency:	69.1%		
Stresses:			
Bending Stress on Gear:			
Enter: Lewis form factor, $y =$	0.100	→	
Normal circular pitch:	0.522 in		
Dynamic factor: $K_v =$	0.979		
Bending stress on gear:	9400 psi [Using effective gear face width]		
Allowable stresses-Bronze: Manganese = 17000 psi; Phosphor = 24000 psi			
Surface Durability: [Hardened steel worm; bronze gear]			
Type of bronze/ $D_g \rightarrow$	>2.5 in	<2.5 in	>8 in
	<8 in	>26 in	<25 in
Sand cast: $C_s =$	857	1000	
Chill cast or forged: $C_s =$			
Centrifugally cast: $C_s =$			
Enter: Materials factor: $C_p =$	857	Sand cast	
Gear Ratio: $m_g =$	6 to 20	20 to 76	Actual $m_g = 30$
Ratio correction factor: $C_m =$	0.759	0.824	0.951
Enter: $C_m =$	0.824		
Sliding velocity:	<700	700 - 3000	>3000
Velocity factor: $C_v =$	0.466	0.498	0.763
Enter: $C_v =$	0.466		
Rated tangential load: $W_R =$	1193 lb		
Must be $> W_f =$	480 lb		
OK For sand cast bronze			
Can use Manganese bronze based on bending stress in gear.			
Worm diameter is too large.			

Wormgearing - Design		Problem 10-24B	
Input Data:			
Desired output torque	$T_d = 1200$ lb-in		
Output speed	$n_g = 20$ rpm		
Velocity Ratio	$VR = 30$		
Design Decisions:			
Diametral pitch	$P_d = 10$		
No. of worm threads	$N_w = 2$		
Required No. of gear teeth:	$N_g = 60$		
Specify No. of gear teeth:	$N_g = 60$		
Normal pressure angle	$\phi_n = 14.5$ degrees		
Computed Results and Additional Inputs:			
Actual input speed:	$n_w = 600$ rpm		
Actual velocity ratio:	$VR = 30$		
Gear pitch diameter:	$D_g = 6$ in		
Specify worm diameter	$D_w = 1.250$ in		
Actual center distance:	$C = 3.625$ in		
	$C^{0.875}/D_w = 2.47$ OK		
	Should be >1.6 and <3.0		
Circular pitch of gear:	$p_g = 0.314$ in		
Axial pitch of worm:	$p_{xw} = 0.314$ in		
Lead of the worm:	$L = 0.628$ in		
Lead angle:	$\lambda = 9.090$ deg		
Addendum:	$a = 0.100$ in		
Dedendum:	$b = 0.116$ in		
Worm outside diameter:	$D_{ow} = 1.450$ in		
Worm root diameter:	$D_{rw} = 1.019$ in		
Nominal worm face length:	$F_{whom} = 2.191$ in		
Gear throat diameter:	$D_{ts} = 6.200$ in		
Nominal gear face width:	$F_{eg} = 0.735$ in		
Max effective gear face width:	$0.67 \cdot D_w = 0.838$ in		
Effective gear face width:	$F_e = 0.625$ in		
Given face width is small; Could use $F > 0.735$ [Used given face width]			
Additional Computed Results:			
Pitch line speed - Gear:	31.42 ft/min		
Sliding velocity $v_s =$	199 ft/min		
Coefficient of friction:	0.043 If $v_s > 10$ ft/min		
Forces: (lb)			
	Gear	Worm	
Tangential:	400	82	
Radial:	106	106	
Axial:	82	400	
Friction force, $W_f =$	18.3 lb		
Power:			
Power output from gear:	0.381 hp		
Power loss - friction:	0.110 hp		
Power input:	0.491 hp		
Efficiency:	77.6%		
Stresses:			
Bending Stress on Gear:			
Enter Lewis form factor y	0.103		
Normal circular pitch:	0.310 in		
Dynamic factor: $K_v =$	0.974		
Bending stress on gear:	21171 psi [Using effective gear face width]		
Allowable stresses-Bronze: Manganese = 17000 psi; Phosphor = 24000 psi			
Surface Durability: [Hardened steel worm; bronze gear]			
Type of bronze/ $D_g \rightarrow$	>2.5 in	<2.5 in	>8 in
	<8 in	>8 in	>25 in
Sand cast: $C_s =$	819	1000	
Chill cast or forged: $C_s =$		1057	1000
Centrifugally cast: $C_s =$			1111
Enter Materials factor $C_m =$	819	Sand cast	
Gear Ratio: $m_g =$	6 to 20	20 to 76	>76
	Actual $m_g = 30$		
Ratio correction factor: $C_m =$	0.759	0.824	0.951
Enter $C_m =$	0.824		
Sliding velocity:	<700	700 - 3000	>3000
	Actual $v_s = 199$		
Velocity factor: $C_v =$	0.530	0.648	1.090
Enter $C_v =$	0.530		
Rated tangential load: $W_{tR} =$	937 lb		
Must be $> W_t =$	400 lb		
OK For sand cast bronze			
Can use Phosphor bronze based on bending stress in gear.			

Wormgearing - Design		Problem: 10-21A Revised	
Input Data: Desired output torque: $T_o = 1200$ lb-in Output speed: $n_o = 20$ rpm Velocity Ratio: $VR = 30$			
Design Decisions: Diametral pitch: $P_d = 6$ No. of worm threads: $N_w = 1$ Required No. of gear teeth: $N_g = 30$ Specify No. of gear teeth: $N_g = 30$ Normal pressure angle: $\phi_n = 14.5$ degrees			
Computed Results and Additional Inputs: Actual input speed: $n_w = 600$ rpm Actual velocity ratio: $VR = 30$ Gear pitch diameter: $D_g = 5$ in Specify worm diameter: $D_w = 1.750$ in Actual center distance: $C = 3.375$ in $C^{0.875}/D_w = 1.66$ OK Should be >1.6 and <3.0			
Circular pitch of gear: $p_g = 0.524$ in Axial pitch of worm: $p_{aw} = 0.524$ in Lead of the worm: $L = 0.524$ in Lead angle: $\lambda = 5.440$ deg Addendum: $a = 0.167$ in Dedendum: $b = 0.193$ in Worm outside diameter: $D_{ow} = 2.083$ in Worm root diameter: $D_{rw} = 1.364$ in Nominal worm face length: $F_{whom} = 2.582$ in Gear throat diameter: $D_{tg} = 5.333$ in Nominal gear face width: $F_{eg} = 1.130$ in Max effective gear face width: $0.67 \cdot D_w = 1.173$ in Effective gear face width: $F_e = 1.130$ in [Used nominal face width]			
Additional Computed Results: Pitch line speed - Gear: 26.18 ft/min Sliding velocity $v_s = 276$ ft/min Coefficient of friction: 0.038 If $v_s > 10$ ft/min			
Forces: (lb) Gear Worm Tangential: 480 65 Radial: 125 125 Axial: 65 480 Friction force, $W_f = 19.0$ lb			
Power: Power output from gear: 0.381 hp Power loss - friction: 0.159 hp Power Input: 0.540 hp			
Efficiency: 70.6%			
Stresses:			
Bending Stress on Gear: Enter Lewis form factor, $y = 0.100 \rightarrow$ Normal circular pitch: 0.521 in Dynamic factor: $K_v = 0.979$			
Bending stress on gear: 8324 psi [Using effective gear face width] Allowable stresses-Bronze: Manganese = 17000 psi; Phosphor = 24000 psi			
Surface Durability: [Hardened steel worm; bronze gear] Type of bronze/ $D_g \rightarrow$ >2.5 in <2.5 in >8 in <8 in >25 in <25 in Sand cast: $C_s = 857$ 1000 Chill cast or forged: $C_s =$ 1093 1000 Centrifugally cast: $C_s =$ 1126 1000 Enter Materials factor: $C_m = 857$ Sand cast			
Gear Ratio: $m_g = 6$ to 20 20 to 76 >76 Actual $m_g = 30$ Ratio correction factor: $C_m = 0.759$ 0.824 0.951 Enter $C_p = 0.824$			
Sliding velocity: <700 700 - 3000 >3000 Actual $v_s = 276$ Velocity factor: $C_v = 0.486$ 0.537 0.845 Enter $C_v = 0.486$			
Rated tangential load: $W_{tr} = 1406$ lb OK For sand cast bronze Must be $> W_f = 480$ lb			
Can use Manganese bronze based on bending stress in gear. Changed worm diameter to 1.75 in; Changed face width to 1.13 in			

CHAPTER 11 KEYS, COUPLINGS, AND SEALS

GENERAL NOTES FOR KEY DESIGN PROBLEMS 1-15:

FOR SAE 1018 CD STEEL: $S_y = 54,000 \text{ psi}$. IF KEY MATERIAL IS WEAKEST -

$$\tau_d = \frac{0.5 S_y}{N} = \frac{0.5(54,000)}{3} = 9,000 \text{ psi}$$

$$\sigma_d = \frac{S_y}{N} = \frac{54,000}{3} = 18,000 \text{ psi}$$

1. $D_{\text{SHAFT}} = 2.00 \text{ IN}$; USE $\frac{1}{2} \text{ IN}$ SQ. KEY, SAE 1018 CD

$$L = \frac{2T}{\tau_d D W} = \frac{2(21,000 \text{ LB-IN})}{(9,000 \text{ LB/IN}^2)(2.00 \text{ IN})(0.5 \text{ IN})} = 4.667 \text{ IN}$$

BUT HUB LENGTH = 4.00 IN. USE SAE 1045 CD; $S_y = 77,000 \text{ psi}$

$$\tau_d = 0.5 S_y / N = 0.5(77,000) / 3 = 12,833 \text{ psi}$$

$$L = \frac{2(21,000)}{(12,830)(2.00)(0.5)} = 3.27 \text{ IN}; \text{ USE } L = 3\frac{3}{4} = 3.75 \text{ IN.}$$

2. $D_s = 3.60 \text{ IN}$; USE $\frac{7}{8} \text{ IN}$ SQUARE KEY; SAE 1018 CD

$$L = \frac{2(21,000)}{(9,000)(3.60)(0.875)} = 1.48 \text{ IN}$$

USE $L = 3.50 \text{ IN}$ TO MORE NEARLY MATCH HUB LENGTH

3. $D_s = 1.75 \text{ IN}$; USE $\frac{3}{8} \text{ IN}$ SQ. KEY; SAE 1018 CD; $\tau_d = 9,000 \text{ psi}$

FOR HUB; CLASS 20 CI, $S_u = 20,000 \text{ psi}$

FOR BEARING $\sigma_d = \frac{20,000}{3} = 6,667 \text{ psi}$ (CONSERVATIVE)

BECAUSE COMPRESSIVE STRENGTH OF CI IS MUCH GREATER THAN TENSILE STRENGTH.

SHEAR OF KEY: $L = \frac{2T}{\tau_d D W} = \frac{2(1112)}{(9,000)(1.75)(.375)} = 0.377 \text{ IN}$

BEARING ON HUB: $L = \frac{4T}{\sigma_d D H} = \frac{4(1112)}{(6,667)(1.75)(.375)} = 1.02 \text{ IN}$

USE $L = 1.50 \text{ IN}$ TO MATCH HUB LENGTH.

4. $T = 63,000(110) / 1700 = 4,076 \text{ LB-IN}$; $D_s = 2.50 \text{ IN}$; USE $\frac{5}{8} \text{ SQ. KEY}$

$$L = \frac{2T}{\tau_d D W} = \frac{2(4,076)}{(9,000)(2.50)(0.625)} = 0.580 \text{ IN}$$

USE $L = 2.50 \text{ IN}$ TO MORE NEARLY MATCH HUB LENGTH.

5.

EXPRESS DATA FROM TABLE 11-6 AS $T = K D^2 L$
REQ'D $K = T / D^2 L$ ($L = \text{HUB LENGTH}$) USE B-FIT

a) PROB. 1 DATA: $T = 21000$; $D = 2.00 \text{ IN}$, $L = 4.00 \text{ IN}$.

$$K = \frac{21000}{(2.00)^2 (4.0)} = 1313 \text{ TOO HIGH FOR ANY SPLINE IN TABLE 11-6}$$

b) PROB. 2 DATA: $T = 21000 \text{ LB-IN}$; $D = 3.60 \text{ IN}$; $L = 4.00 \text{ IN}$.

$$K = \frac{21000}{(3.60)^2 (4.0)} = 405; \text{ USE 16 SPLINES; } K = 521$$

c) PROB. 3 DATA: $T = 1112 \text{ LB-IN}$; $D = 1.75 \text{ IN}$; $L = 1.75 \text{ IN}$

$$K = \frac{1112}{(1.75)^2 (1.75)} = 208 \text{ USE 6 SPLINES}$$

d) PROB. 4 DATA: $T = 4076 \text{ LB-IN}$; $D = 2.50 \text{ IN}$; $L = 3.25 \text{ IN}$

$$K = \frac{4076}{(2.50)^2 (3.25)} = 201 \text{ USE 6 SPLINES}$$

6.

$$\text{AT } 220 \text{ HP: } T_2 = 63000(220)/1700 = 8153 \text{ LB-IN}$$

$$\text{AT } 110 \text{ HP: } T_1 = 63000(110)/1700 = 4076 \text{ LB-IN}$$

PIN SHOULD SHEAR AT T_2 ; $T_2 = S_{MS} = S_M(0.75)$ (SECT. 2-2)

PIN SHOULD NOT YIELD AT T_1 ; $T_1 \leq S_{YS} = S_Y(0.5)$

FOR A GIVEN PIN d AND SHAFT D IN EQ. 11-18

$$S_{SY} = T_1 = \frac{4 T_1}{D \pi d^2}; \quad T_2 = \frac{4 T_2}{D \pi d^2} = 0.75 S_M$$

$$\text{RATIO } \frac{0.5 S_Y}{0.75 S_M} = \frac{\frac{4 T_1}{D \pi d^2}}{\frac{4 T_2}{D \pi d^2}} = \frac{T_1}{T_2}; \quad S_Y = \frac{0.75 T_1 S_M}{0.5 T_2}$$

$$\text{MIN. } S_Y = \frac{0.75(4076) S_M}{0.5(8153)} = 0.75 S_M \quad \text{MOST COLD DRAWN STEELS HAVE } S_Y \geq 0.75 S_M$$

$$\text{FOR SAE 1018 CD. } d = \sqrt{\frac{4 T_2}{D \pi T_2}} = \sqrt{\frac{4(8153)}{(2.50)(\pi)(1.15)(64000)}} = 0.294 \text{ in.}$$

$$S_Y = 54 \text{ KSI}; \quad S_M = 64 \text{ KSI}$$

$$\text{AT } T_1 = 4076 \text{ LB-IN}; \quad T = \frac{4(4076)}{(2.50)(\pi)(0.294)^2} = 24009 \text{ psi} < \frac{S_Y = 54000 \text{ PSI}}{2} = 27000 \text{ PSI}$$

7. FROM EX. PROB. 12-3: $T = 4168 \text{ LB}\cdot\text{IN}$; $D_2 = 2\frac{1}{4} \text{ IN}$; $D_5 = 1\frac{1}{2} \text{ IN}$.

SPROCKET: $\frac{1}{2} \text{ IN}$ SQUARE KEY; SAE 1018 CD

$$L = \frac{2T}{T_s DW} = \frac{2(4168)}{(9000)(2.25)(0.5)} = 0.823 \text{ IN} \quad \text{USE } L = 1.00 \text{ IN}$$

WORM GEAR: $\frac{3}{8} \text{ SQ. KEY}$

$$L = \frac{2(4168)}{(9000)(1.50)(0.375)} = 1.647 \text{ IN.} \quad \text{USE } L = 1.75 \text{ IN.}$$

8.

WOODRUFF KEY 204: NOMINAL $W = \frac{2}{32} = \frac{1}{16} \text{ IN}$; NOMINAL $B = \frac{4}{8} = \frac{1}{2} \text{ IN}$
ACTUAL DIMS. IN TABLE 11-3

9.

WOODRUFF KEY 1628: NOM. $W = \frac{16}{32} = \frac{1}{2} \text{ IN}$; NOM. $B = \frac{28}{8} = 3\frac{1}{2} \text{ IN}$

10, 11, 12 ARE DRAWINGS

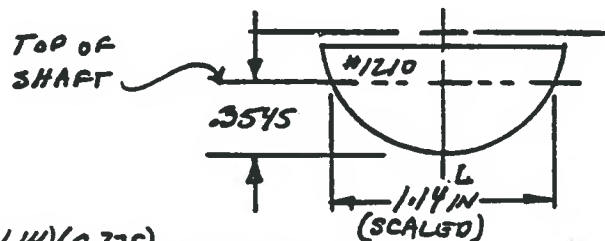
13.

$$T = F D / 2 = F = 2T / D$$

$$\text{KEY } W = \frac{3}{8}$$

$$A_s = LW; T = \frac{F}{A_s} = \frac{2T}{DLW}$$

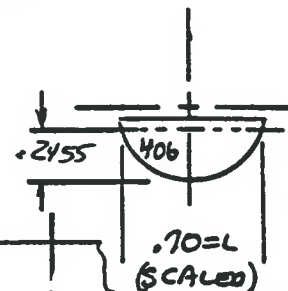
$$T = \frac{T_s DLW}{2} = \frac{(9000)(1.500)(1.14)(0.375)}{2} = 2886 \text{ LB}\cdot\text{IN}$$



14.

$$T = \frac{T_s DLW}{2} = \frac{(9000)(.500)(.70)(.125)}{2} = 197 \text{ LB}\cdot\text{IN}$$

$$\text{KEY } W = \frac{1}{8} \text{ IN}$$



15.

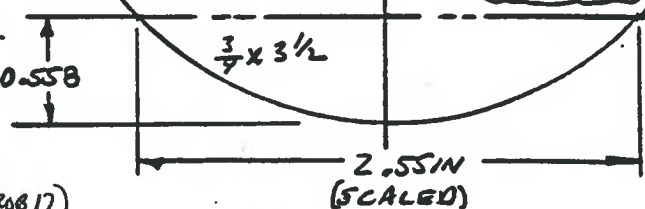
$$T = \frac{(9000)(3.25)(2.55)(0.75)}{2} = 27,970 \text{ LB}\cdot\text{IN}$$

19.

a) $T = 1390^2 = 139(1.50)^2$ (PROB 16) 0.558
 $= 313 \text{ LB}\cdot\text{IN} / \text{IN OF HUB L.}$

b) $T = 3260^2 = 326(3.50)^2 =$
 $= 3994 \text{ LB}\cdot\text{IN} / \text{IN OF HUB L.}$ (PROB 17)

c) $T = 688 D^2 = 688(2.500)^2 = 4300 \text{ LB}\cdot\text{IN} / \text{IN. OF HUB L.}$ (PROB 18)



NOTE: Problems 20-46 call for narrative answers for which the proper information can be found in the text. Guidance is provided below for sections in which additional information can be found.

20. Section 11-6 includes discussion of applying set screws to transmit torque. A table of approximate holding force capacity vs. set screw size is provided.
21. Press fit is described briefly in Section 11-6. More discussion follows in Chapter 13.
22. Section 11-7 describes both rigid and flexible couplings and compares their performance. Examples of commercially available couplings are shown and described.
23. Section 11-8 contains general information about universal joints.
24. Section 11-9 contains general information about retaining rings and other means of locating machine elements axially on shafts and in other devices. Included are collars, shoulders, spacers, and locknuts.
25. to 38. Section 11-10 contains general information about seals.
39. to 46. Section 11-11 contains general information about seal materials, including elastomers.
 40. to 45. A list of 14 elastomers is included in Section 11-11. Following the list of 14, elastomers, their general performance capabilities are described.
 46. The required conditions for shafts on which elastomeric seals operate are discussed in the last part of Section 11-11. Examples are:
 - Steels, hardened to HRC 30 with tolerances of less than ± 0.005 in (0.13 mm) are typically used for shafts on which seals operate. The surface must be free of burrs with a surface finish of 10 to 20 microinches is recommended. Lubrication is recommended.

CHAPTER 12

SHAFT DESIGN

GENERAL NOTES CONCERNING SOLUTIONS TO SHAFT DESIGN PROBLEMS

- Design values for stress concentrations as given in Section 12-4 are used for the initial calculations. These values must be checked once final design details are specified for diameters, fillet radii, and other features.
- Estimates are originally used for the size factors used in calculations because they depend on the shaft sizes that are unknown at the start of a design problem. These values must be checked once final design decisions have been specified.
- The choice of the reliability factor, C_R , is a design decision. Other values may be preferred.
- In most cases, the proposed final values for diameters are expected to be safe because trial values are typically conservative and because final specified diameters are typically made to the next larger preferred size according to Appendix Table A-2.
- Final specifications for diameters where bearings are to be mounted must await the selections for suitable bearings that can accommodate the radial and thrust loads applied to them. This process is described in Chapter 14 and the MDESIGN – MOTT software is an excellent tool for making those decisions. The computed 'minimum required diameter' from the shaft design process should be used as to limit the bearing selection to only feasible sizes.

Shafts with Only Radial Loads Applied to Them:

Problems P1 through P30 relate to one of the Figures P12-1 through P12-17 showing shafts carrying a variety of combinations of gears, belt sheaves, chain sprockets, and a few other items such as a flywheel and a propeller-type fan. All of these elements apply only radial loads to the shafts on which they are mounted.

- **Problems 1-11 include only forces and torques exerted by gears on shafts. No separate solutions for these problems are included here.**
- **Problems 12-21 include only forces and torques exerted by belt drives and chain drives on shafts. No separate solutions are included here.**
- **Problems 22-30 are comprehensive design problems that use the same shaft assemblies that are used for Problems 1-21. The solutions to these problems include the analyses of forces and torques and should be used as the solutions for problems 1-21.**
- **The parts of the solutions for torques and forces give discrete, single-answers.**
- **The remaining parts of the comprehensive shaft designs include many design decisions and multiple solutions are possible. The given solutions should be considered examples only.**

There are multiple ways in which the problems P1 through P30 may be assigned. The following table may help instructors decide how to assign the problems for student solution and may help students

comprehend how the sets of problems lead to the more general shaft design. Any combination of problems may be chosen.

<u>Torques and Forces Acting Radial to Shaft</u>	<u>Comprehensive</u>
Figure P12-1: P1 – Gear <i>B</i> ; P14 – Sheave <i>D</i>	P22
Figure P12-2: P2 – Gear <i>C</i> ; P12 – Sprocket <i>D</i> ; P13 – Pulley <i>A</i>	P23
Figure P12-3: P3 – Gear <i>B</i> ; P15 – Sprocket <i>C</i> ; P16 – Sheaves <i>D, E</i>	P24
Figure P12-4: P4 – Gear <i>A</i> ; P19 – Sprockets <i>C, D</i>	P25
Figure P12-5: P5 – Gear <i>D</i> ; P20 – Sheave <i>A</i> ; P21 – Sprocket <i>E</i>	P26
Figure P12-6: P6 – Gear <i>E</i> ; (No separate analysis of Sheave <i>A</i>)	P27 (Includes Sheave <i>A</i>)
Figure P12-7: P7 – Gear <i>C</i> ; P8 – Gear <i>A</i>	P28 (Includes Sheaves <i>D, E</i>)
Figure P12-9: P9 – Gear <i>C</i> ; P10 – Gear <i>D</i> ; P11 – Gear <i>F</i>	P29 (Includes Sheave <i>B</i>)
Figure P12-17: P17 – Sheave <i>C</i> ; P18 – Pulley <i>D</i>	P30 (Includes Fan <i>A</i>)

Shafts with both Radial and Axial Loads Applied to Them:

Problems P31 to P34 deal with shafts carrying helical gears and wormgears that produce forces directed axially in addition to radial forces. Solutions are only shown for the comprehensive problems (12-32 and 12-34) in which the details of the analyses of torques and forces are included.

<u>Torques and Forces Acting Radial and Axial to Shaft</u>	<u>Comprehensive</u>
Figure P12-31: P31 – Helical Gear <i>B</i>	P32
Figure P12-33: P33 – Wormgear <i>C</i>	P34 (Includes Sheave <i>A</i>)

Other Comprehensive Design Problems

Problems 35 to 41 contain a variety of loading situations for which the general solution procedure must be adapted. Some of the problems involve more than one shaft, considering shafts for mating gears and multiple reductions.

Figure P12-35: P35 – Double reduction helical drive

Figure P10-8 in Chapter 10: P36 – Bevel gear drive

Figure P12-37: P37 – Bevel gear drive with two chain sprockets

Figure P12-38: P38 – Double reduction spur gear drive; design three shafts.

Figure P12-39: P39 – Drive system consisting of an electric motor, a V-belt drive, a double reduction spur gear type reducer, and a chain drive.

Figure P12-40: P40 – Shaft with three spur gears

Figure P12-41: P41 – Shaft for windshield wiper mechanism with two levers

FIGURE P12-1

- 22 TORQUE ON GEAR B: $T_B = 63000(30)/550 = 3436 \text{ LB-IN}$
SAME TORQUE ON SHEAVE D: $T_D = 3436 \text{ LB-IN}$
TORQUE IN SHAFT: $T_{A-B} = 0$; $T_{B-D} = 3436 \text{ LB-IN}$
FORCES ON GEAR B:

$$W_{tB} = \frac{T_B}{R_B} = \frac{3436 \text{ LB-IN}}{8 \text{ IN}} = 430 \text{ LB} \leftarrow = F_{Bx}$$

$$W_{rB} = t_B \tan 20^\circ = 156 \text{ LB} \downarrow = F_{By}$$

FORCES ON SHEAVE D

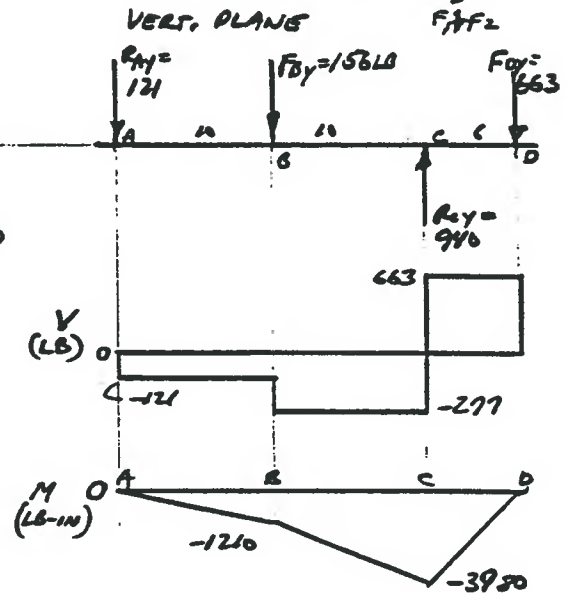
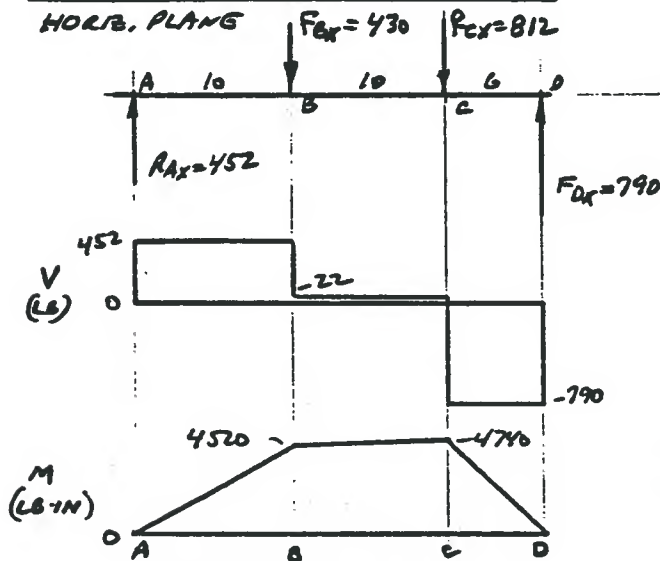
$$F_1 - F_2 = \frac{T_D}{R_D} = \frac{3436 \text{ LB-IN}}{5.0 \text{ IN}} = 687 \text{ LB} \nearrow 40^\circ$$

$$F_1 + F_2 = 1.5(F_1 - F_2) = 1.5(687) = 1031 \text{ LB} = F_D$$

$$F_{Dx} = F_D \cos 40^\circ = 790 \text{ LB} \rightarrow$$

$$F_{Dy} = F_D \sin 40^\circ = 663 \text{ LB} \downarrow$$

BENDING MOMENT DIAGRAMS



$$M_B = \sqrt{4520^2 + 1210^2} = 4679 \text{ LB-IN}$$

$$M_C = \sqrt{4740^2 + 3980^2} = 6189 \text{ LB-IN}$$

$$T_C = 3436 \text{ LB-IN}$$

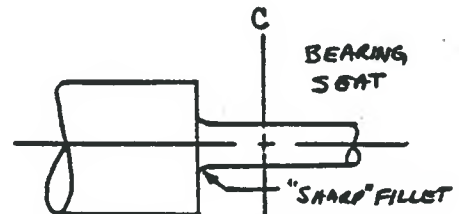
ASSUME $K_t = 2.5$ AT LEFT OF C

SAE 1040 CD; $S_u = 80 \text{ KSI}$; $S_y = 71 \text{ KSI}$

$S_m = 30 \text{ KSI}$ (FIG. 5-8 MACHINED) $C_R = 0.81$, $C_s = 0.80$

$$S_m' = C_R C_s S_m = (0.80)(0.81)(30) = 19440 \text{ PSI}$$

$$D_C = \left[\frac{22(3)}{\pi} \sqrt{\left[\frac{2.5(6189)}{19440} \right]^2 + \frac{3}{4} \left[\frac{3436}{71000} \right]^2} \right]^{1/3} = 2.90 \text{ IN.}$$



DESIGN GEOMETRY AT C

USE $N = 3$

SPECIFY $D = 3.00 \text{ IN.}$
 $C_s = 0.78$ OK

23

FIGURE P12-2

TORQUE ON PULLEY A: $T_A = 63000(\text{lb})/200 = 3150 \text{ LB-IN}$ " ON GEAR C: $T_C = 63000(\text{lb})/240 = 1890 \text{ LB-IN}$ " ON SPROCKET D: $T_D = 63000(4)/200 = 1260 \text{ LB-IN}$ TORQUE DISTRIBUTION IN SHAFT: $T_{A-C} = 3150 \text{ LB-IN}$; $T_{C-D} = 1260 \text{ LB-IN}$; $T_{D-E} = 0$

FORCES ON PULLEY A:

$$F_1 - F_2 = T_A / R_A = 3150 / 10 = 315 \text{ LB}$$

$$F_1 + F_2 = 2.0(F_1 - F_2) = 2.0(315) = 630 \text{ LB} \downarrow = F_{Ay}; F_{Ax} = 0$$

FORCES ON GEAR C:

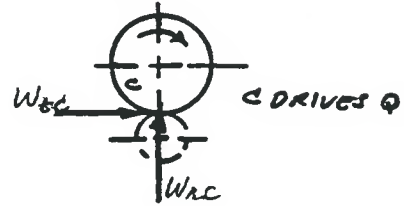
$$W_{tc} = T_C / R_c = \frac{1890 \text{ LB-IN}}{5.0 \text{ IN}} = 378 \text{ LB} \rightarrow = F_{cx}$$

$$W_{nc} = W_{tc} \tan 20^\circ = 138 \text{ LB} \uparrow = F_{cy}$$

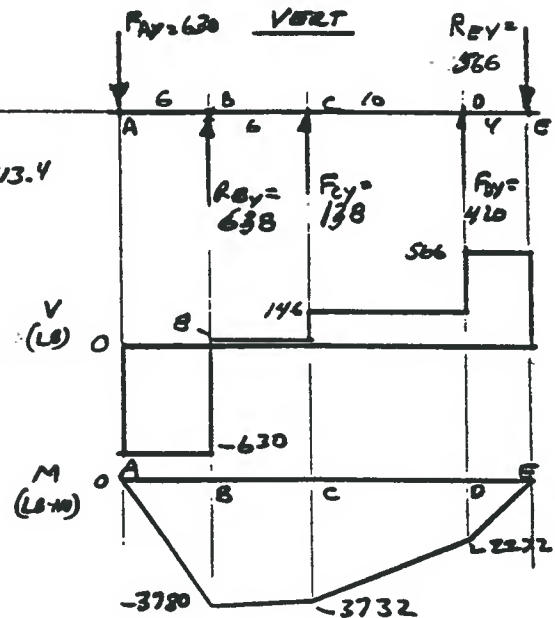
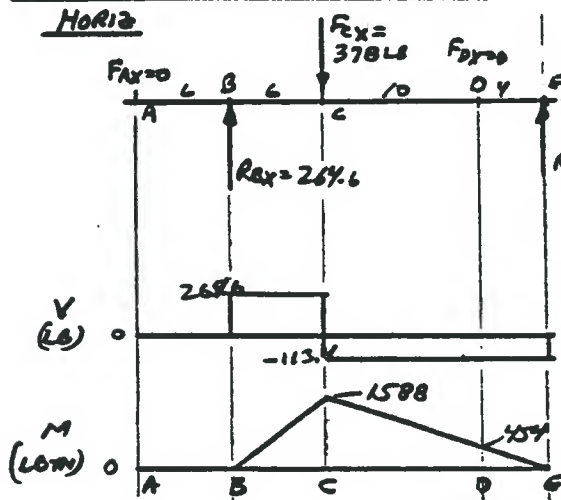
FORCES ON SPROCKET D:

$$F_1 = F_D = F_{Dy} = \frac{T_D}{R_D} = \frac{1260}{3} = 420 \text{ LB} \uparrow$$

$$F_{Dx} = 0$$



BENDING MOMENT DIAGRAMS



$$M_B = 3780 \text{ LB-IN}$$

$$M_C = \sqrt{1588^2 + 3732^2} = 4056 \text{ LB-IN}$$

$$M_D = \sqrt{454^2 + 2272^2} = 2317 \text{ LB-IN}$$

$$T_C = 3150 \text{ LB-IN TO LEFT}; K_t = 3.0 \text{ RING GEAR}$$

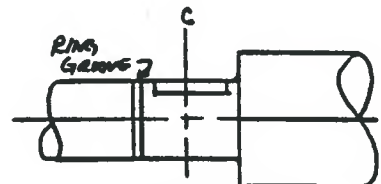
$$T_D = 1260 \text{ LB-IN TO RIGHT}; K_t = 2.0 \text{ KEY}$$

$$\text{SAE 1117 CD}; S_u = 69 \text{ KSI}; S_y = 57 \text{ KSI}$$

$$S_m = 28 \text{ KSI}; S_m' = C_s C_G S_u = (0.85)(0.81)(28) = 17.9 \text{ KSI}$$

$$D_c = \left[\frac{32(3)}{\pi} \sqrt{\left(\frac{3.0(4056)}{17900} \right)^2 + \frac{3}{4} \left(\frac{3150}{51000} \right)^2} \right]^{1/3} = 2.75 \text{ IN.}$$

$$\text{INCREASE BY 6\% AT GROOVE } D \approx 1.06(2.75) = 2.92 \text{ IN.}$$



DESIGN AT C

SPECIFY $D_c = 3.00 \text{ IN}$ $C_s = 0.78$ OK

FIGURE P12-3

24

TORQUE ON GEAR B: $T_B = 63000(5)/480 = 656 \text{ LB}\cdot\text{IN}$

" ON SHEAVES D AND E: $T_D = T_E = 63000(3)/480 = 394 \text{ LB}\cdot\text{IN}$

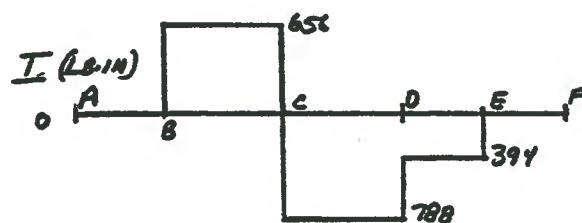
" ON SPROCKET C: $T_C = 63000(11)/480 = 1444 \text{ LB}\cdot\text{IN}$

TORQUE DISTRIBUTION IN SHAFT:

$T_{AB} = 0$; $T_{BC} = 656 \text{ LB}\cdot\text{IN}$

$T_{CD} = 788 \text{ LB}\cdot\text{IN}$

$T_{DE} = 394 \text{ LB}\cdot\text{IN}$; $T_{EF} = 0$



FORCES ON GEAR B:

$W_{tB} = T_B / r_B = 656 / 1.5 = 437 \text{ LB} \rightarrow = F_{Bx}$

$W_{nB} = W_{tB} \tan 20^\circ = 159 \text{ LB} \uparrow = F_{By}$

FORCES ON SPROCKET C: $F_C = T_C / r_C = 1444 / 5.0 = 289 \text{ LB}$

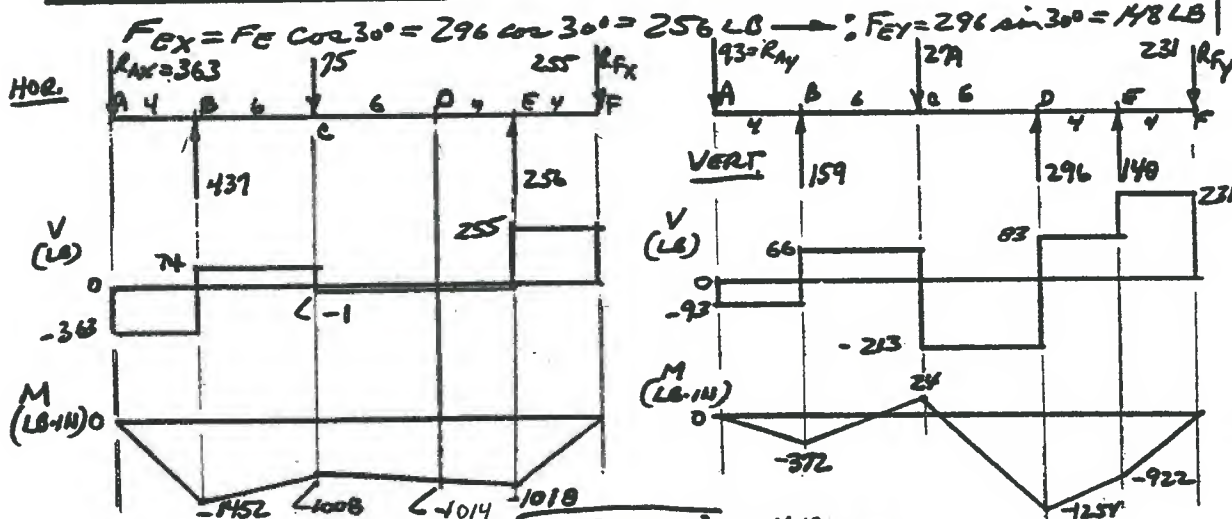
$F_{Cx} = F_C \sin 15^\circ = 289 \sin 15^\circ = 75 \text{ LB} \rightarrow$

$F_{Cy} = F_C \cos 15^\circ = 289 \cos 15^\circ = 279 \text{ LB} \downarrow$

FORCES ON SHEAVE D: $F_1 = F_2 = T_D / r_D = 394 / 2.0 = 197 \text{ LB}$

$F_D = F_1 + F_2 = 1.5(F_1 - F_2) = 1.5(97) = 296 \text{ LB} \uparrow = F_{Dy}$; $F_{Dx} = 0$

FORCES ON SHEAVE E: $F_E = F_D = 296 \text{ LB}$



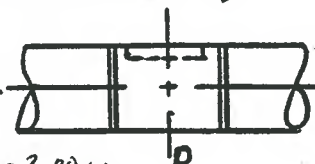
WORST CASE AT D: $M_D = \sqrt{1014^2 + 1254^2} = 1613 \text{ LB}\cdot\text{IN}$

$T_D = 788 \text{ LB}\cdot\text{IN}$ (ATO & TO LEFT); $K_t = 3.0$ RST. RING

SAE 1137 OQT 1300; $S_u = 87 \text{ KSI}$; $S_y = 60 \text{ KSI}$; $S_m = 33 \text{ KSI}$ (FIG 5-8)

$S_m' = 0.9(0.81)(33) = 24.1 \text{ KSI}$ ($C_s = 0.80$ EST.)

$$D_D = \left[\frac{32(3)}{\pi} \sqrt{\left(\frac{3.0(1613)}{24100} \right)^2 + \frac{3}{4} \left(\frac{788}{60000} \right)^2} \right]^{1/3} = 1.83 \text{ IN.}$$



INCREASE BY 6%; $D_D = 1.06(1.83) = 1.94 \text{ IN.}$; SPECIFY $D_D = 2.00 \text{ IN.}$

CHECK: $C_s = 0.81$; $S_m' = 21700 \text{ PSI}$; $D_{D \text{ REQ}} = 2.01 \text{ IN.}$; ACCEPTABLE TO USE $D_D = 2.00 \text{ IN.}$

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FIGURE P12-4

TORQUE ON GEAR A: $T_A = 63000(40)/120 = 21000 \text{ LB}\cdot\text{IN}$

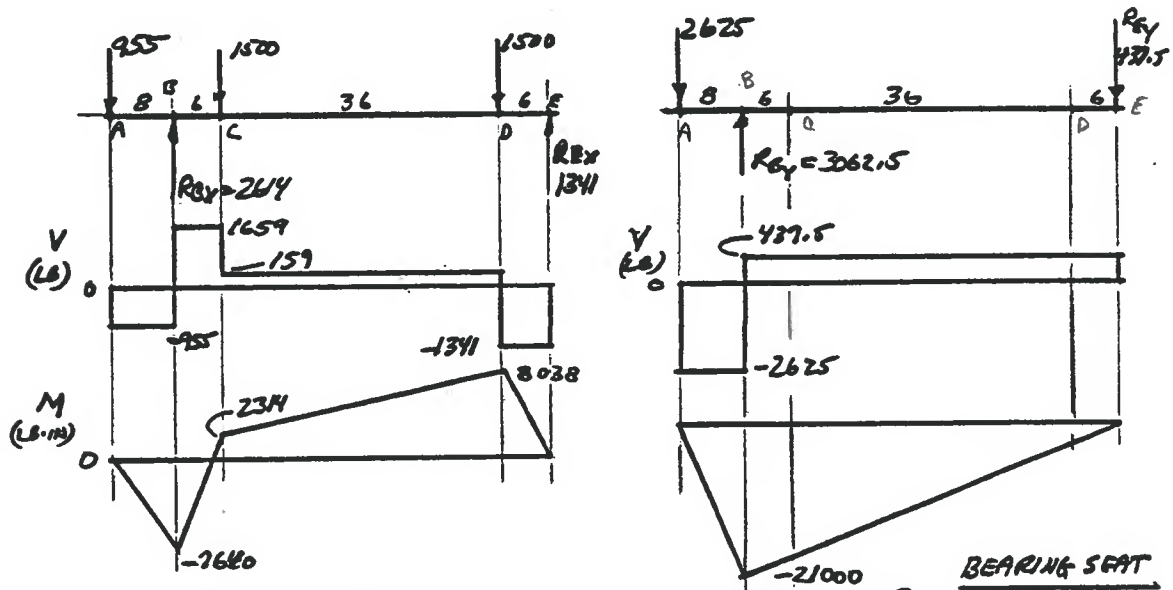
" " ON SPROCKETS C & D: $T_C = T_D = 63000(20)/120 = 10500 \text{ LB}\cdot\text{IN}$

TORQUE IN SHAFT: $T_{A-C} = 21000 \text{ LB}\cdot\text{IN}$; $T_{C-D} = 10500 \text{ LB}\cdot\text{IN}$; $T_{D-E} = 0$

FORCES ON GEAR A: $W_{CA} = T_A/R_A = 21000/8 = 2625 \text{ LB} = F_{AY}$

$W_{HA} = W_{CA} \tan 20^\circ = 955 \text{ LB} \leftarrow = F_{AX}$

FORCES ON SPROCKETS C & D: $F_{CX} = F_{DX} = T/R = 10500/7 = 1500 \text{ LB} \leftarrow$



AT B: $T = 21000 \text{ LB}\cdot\text{IN}$

$$M = \sqrt{7640^2 + 21000^2} = 22347 \text{ LB}\cdot\text{IN}$$

TO RIGHT OF B: $K_t = 2.5$ - SMALL FILLET RAD.

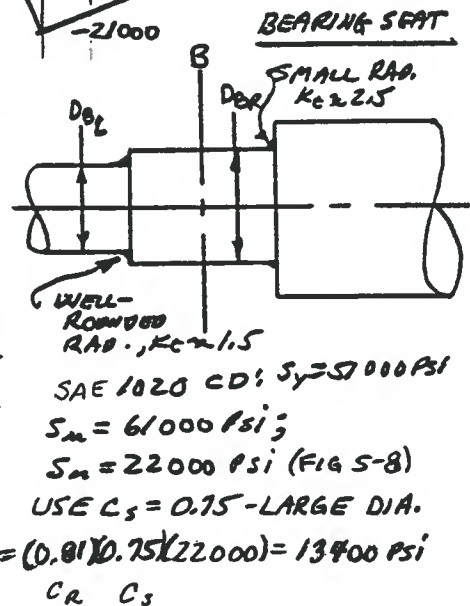
$$D_{BR} = \left[\frac{32(3)}{\pi} \sqrt{\left(\frac{2.5(22347)}{13400} \right)^2 + \frac{3}{4} \left(\frac{21000}{51000} \right)^2} \right]^{1/3}$$

$D_{BR} = 5.05 \text{ IN.}$ SPECIFY 5.20 IN $C_s = 0.73$ OK

TO LEFT OF B: $K_t = 1.5$ - WELL RND. FILLET

$$D_{BL} = \left[\frac{32(3)}{\pi} \sqrt{\left(\frac{1.5(22347)}{13400} \right)^2 + \frac{3}{4} \left(\frac{21000}{51000} \right)^2} \right]^{1/3}$$

$D_{BL} = 4.27 \text{ IN.}$ SPECIFY 4.50 IN $C_s = 0.74$ OK



SAE 1020 CD; $S_y = 57000 \text{ PSI}$

$S_u = 61000 \text{ PSI}$

$S_n = 22000 \text{ PSI (FIG 5-8)}$

USE $C_s = 0.75$ - LARGE DIA.

$$S_n' = (0.81)(0.75)(22000) = 13400 \text{ PSI}$$

$C_R \quad C_S$

FIGURE P12-5

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TORQUE ON SHEAVE A: $T_A = 63000(10)/240 = 2625 \text{ LB}\cdot\text{IN} = T_{AD} \text{ IN SHAFT}$

TORQUE ON GEAR D: $T_D = 63000(15)/240 = 3938 \text{ LB}\cdot\text{IN}$

TORQUE ON SPROCKET E: $T_E = 63000(5)/240 = 1313 \text{ LB}\cdot\text{IN} = T_{DE} \text{ IN SHAFT}$

FORCES ON SHEAVE A: $F_1 - F_2 = T_A / r_A = 2625/6 = 438 \text{ LB}$; $F_{AX} = 0$

$F_A = F_{AY} = F_1 + F_2 = 1.5(F_1 - F_2) = 1.5(438) = 657 \text{ LB}$

FORCES ON GEAR D: $W_{tD} = T_D / r_D = 3938/4 = 985 \text{ LB}$

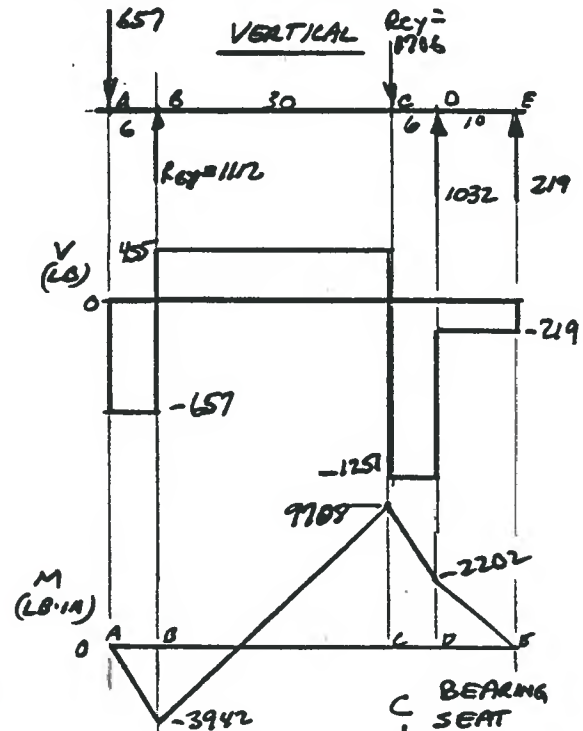
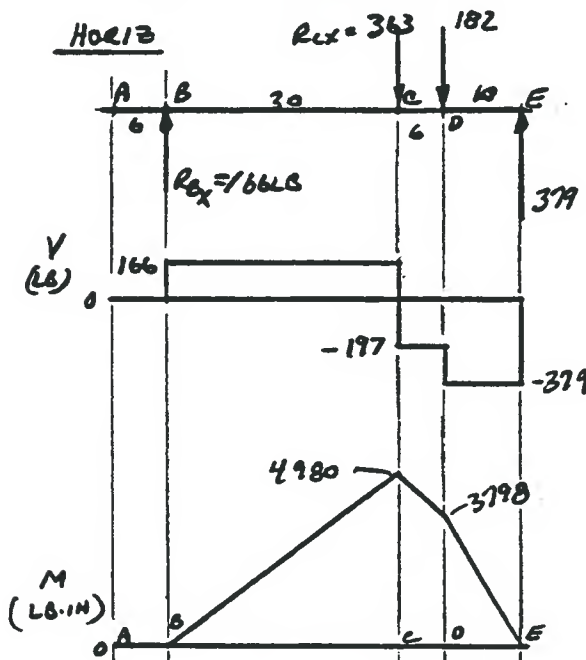
$W_{AD} = W_{tD} \tan 20^\circ = 358 \text{ LB}$

$F_{DX} = 985 \sin 30^\circ - 358 \cos 30^\circ = 182 \text{ LB}$

$F_{DY} = 985 \cos 30^\circ + 358 \sin 30^\circ = 1032 \text{ LB}$

FORCES ON SPROCKET E: $F_E = T_E / r_E = 1313/3 = 438 \text{ LB}$

$F_{EX} = F_E \cos 30^\circ = 379 \text{ LB}$; $F_{EY} = F_E \sin 30^\circ = 219 \text{ LB}$



AT C: $T = 2625 \text{ LB}\cdot\text{IN}$; USE $N = 4$ STRESS

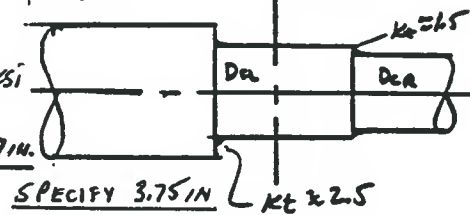
$M = \sqrt{4980^2 + 9709^2} = 10911 \text{ LB}\cdot\text{IN}$

$CH0035 \ 1137 \text{ CD} - S_y = 82000 \text{ PSI}$

$S_m = 98 \text{ KSI}$; $S_u = 37 \text{ KSI}$; $S_m' = 0.8(0.81)(37) = 24.0 \text{ KSI}$

$D_{CL} = \left[\frac{32(Y)}{\pi} \sqrt{\left(\frac{2.5(10911)}{24000} \right)^2 + \frac{3}{4} \left(\frac{2625}{82000} \right)^2} \right]^{1/3} = 3.59 \text{ IN.}$

WITH $K_t = 1.5$, $D_{CR} = 3.03 \text{ IN.}$ SPECIFY 3.20 IN
 $C_s = 0.77 \text{ OK}$



SPECIFY 3.75 IN
 $C_s = 0.76 \text{ OK}$

FIGURE P12-6

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TORQUE ON SHEAVE A AND GEAR E: $T = 63000(20)/300 = 4065 \text{ LB}\cdot\text{IN}$

FORCES ON SHEAVE A: $F_1 - F_2 = T/R_A = 4065/11 = 370 \text{ LB}$

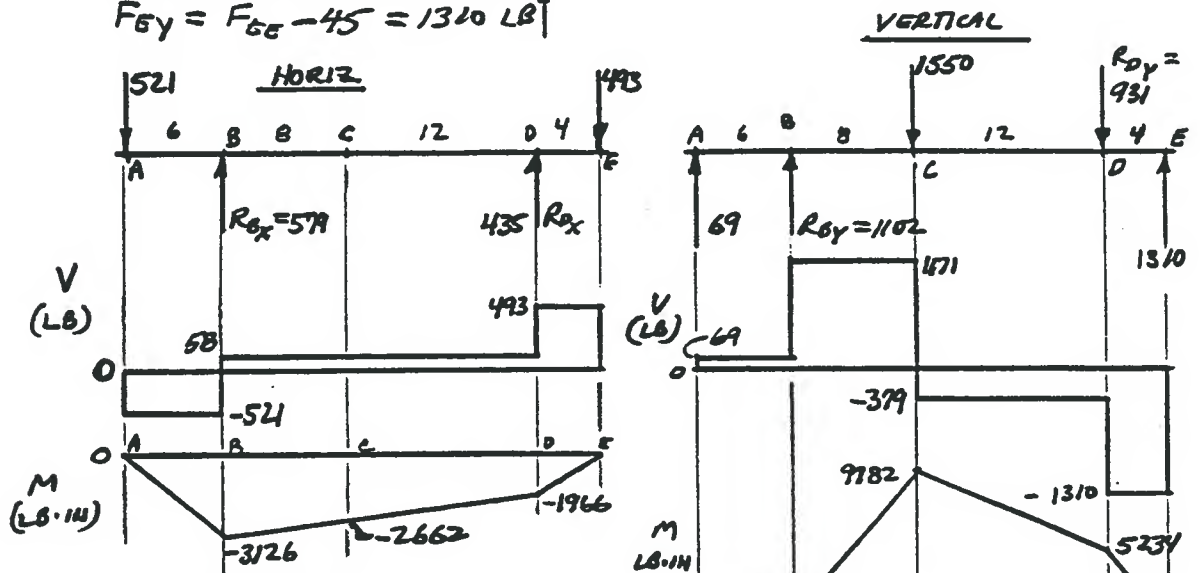
$$F_A = F_1 + F_2 = 1.5(F_1 - F_2) = 554 \text{ LB.}$$

$$F_{Ax} = F_A \cos 20^\circ = 521 \text{ LB} ; F_{Ay} = F_A \sin 20^\circ - 120 \text{ LB} = 189 \text{ LB}$$

FORCES ON GEAR E: $W_{tE} = T/R_E = 4065/3 = 1355 \text{ LB}$

$$W_{tE} = W_{tE} \tan 20^\circ = 493 \text{ LB} \rightarrow = F_{Ex}$$

$$F_{Ey} = F_{tE} - 45 = 1310 \text{ LB}$$



AT C: $T = 4065 \text{ LB}\cdot\text{IN}$

$$M = [2662^2 + 9782^2]^{1/2} = 10138 \text{ LB}\cdot\text{IN}$$

USE $K_t = 2.0$; $N = 3$

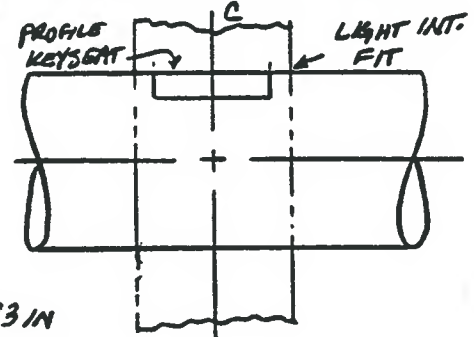
SAE 1050 CD; $S_y = 84 \text{ KSI}$

$S_u = 100 \text{ KSI}$; $S_m = 38 \text{ KSI}$

$$S_m' = 0.8(0.8)(38) = 24.6 \text{ KSI}$$

$$D_c = \left[\frac{32(3)}{\pi} \sqrt{\left(\frac{2.0(10138)}{24600} \right)^2 + \frac{3(4065)}{4(84000)}^2} \right]^{1/3} = 2.93 \text{ IN}$$

SPECIFY $D_c = 3.00 \text{ IN}$ $C_s = 0.78 \text{ OK}$



28

TORQUE ON GEAR A: $T_A = 63000(30)/480 = 3938 \text{ LB}\cdot\text{IN}$

TORQUE ON GEAR C: $T_C = 63000(50)/480 = 6563 \text{ LB}\cdot\text{IN}$

TORQUE ON SHEAVES D AND E: $T_D = T_E = 63000(10)/480 = 1313 \text{ LB}\cdot\text{IN}$

TORQUE IN SHAFT: $T_{AC} = 3938 \text{ LB}\cdot\text{IN}$; $T_{C-D} = 2626 \text{ LB}\cdot\text{IN}$; $T_{D-E} = 1313 \text{ LB}\cdot\text{IN}$
 $T_{E-F} = 0$

(CONTINUED ON NEXT PAGE)

28

(FIGURE P12-7)

(CONTINUED)

FORCES ON GEAR A:

$$W_{tA} = T_A / r_A = 3938 / 2.5 = 1575 \text{ LB} \uparrow = F_{Ay}$$

$$W_{nA} = W_{tA} \tan 20^\circ = 1575 \tan 20^\circ = 573 \text{ LB} \rightarrow = F_{Ax}$$

$$\text{FORCES ON GEAR C: } W_{tC} = T_C / r_C = 6563 / 5 = 1313 \text{ LB} \rightarrow = F_{Cx}$$

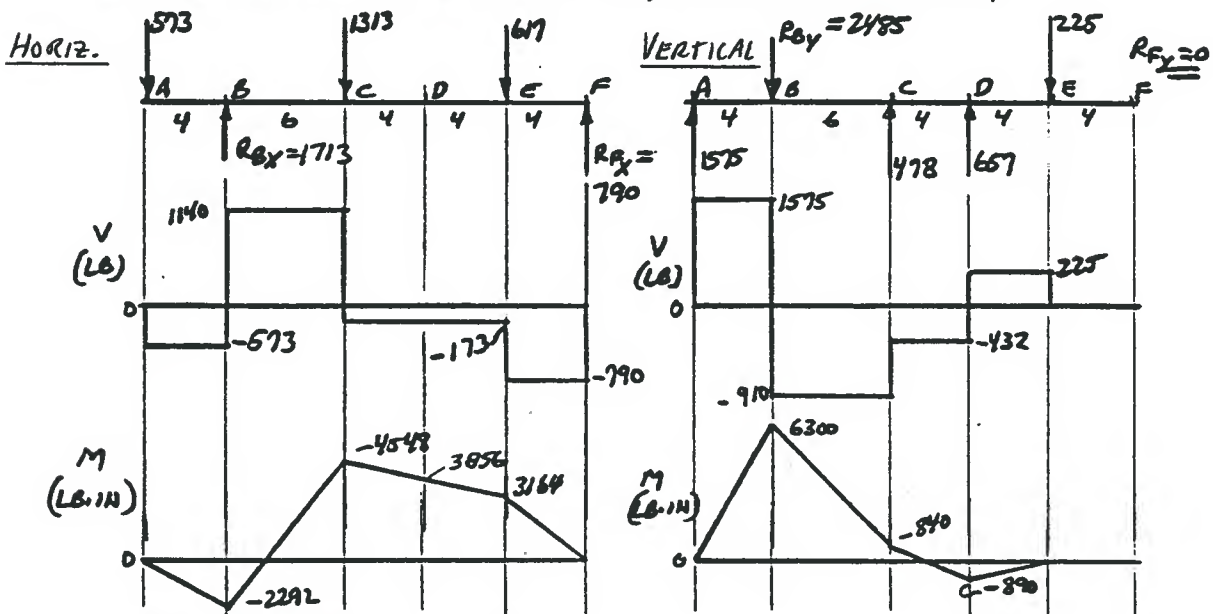
$$W_{nC} = W_{tC} \tan 20^\circ = 1313 \tan 20^\circ = 478 \text{ LB} \uparrow = F_{Cy}$$

$$\text{FORCES ON SHEAVE D: } F_1 - F_2 = T_D / r_D = 1313 / 3 = 438 \text{ LB}$$

$$F_D = F_{Dy} = F_1 + F_2 = 1.5(F_1 - F_2) = 1.5(438) = 657 \text{ LB} \uparrow : F_{Dx} = 0$$

$$\text{FORCES ON SHEAVE E: } F_E = F_D = 657 \text{ LB} \uparrow$$

$$F_{Ex} = F_E \cos 20^\circ = 617 \text{ LB} \rightarrow : F_{Ey} = F_E \sin 20^\circ = 225 \text{ LB} \downarrow$$



$$\text{AT B: } T = 3938 \text{ LB-IN ; } N = 3$$

$$M_B = \sqrt{(2292)^2 + (6300)^2} = 6704 \text{ LB-IN}$$

$$D_{BR} = \left[\frac{32(3)}{\pi} \sqrt{\left(\frac{2.5(6704)}{35,800} \right)^2 + \frac{3}{4} \left(\frac{3938}{133,000} \right)^2} \right]^{1/3} = 2.43 \text{ IN}$$

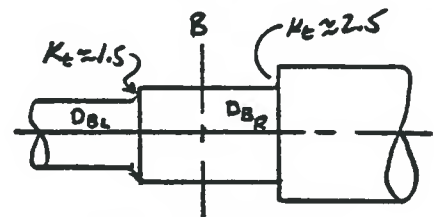
$$D_{BL} = 2.05 \text{ IN WITH } K_t = 1.5$$

SPECIFY D = 2.20 IN

$$C_s = 0.80 \quad \text{OK}$$

$$\text{SPECIFY D} = 2.50 \text{ IN}$$

$$C_s = 0.78 \quad \text{OK}$$



SAE 3140 OQT 1000

$$S_y = 133 \text{ KSI ; } S_m = 152 \text{ KSI}$$

$$S_m = 52 \text{ KSI ; } S_m' = C_s \cdot C_R \cdot S_m$$

$$S_m' = (0.85)(0.81)(52) = 35.8 \text{ KSI}$$

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FIGURE P12-9

TORQUE ON SHAFT B: $T_B = 63000(2.5)/220 = 716 \text{ LB}\cdot\text{IN}$

" ON GEARS C AND F: $T_C = T_F = 63000(5)/220 = 1432 \text{ LB}\cdot\text{IN}$

" ON GEAR D: $T_D = 63000(12.5)/220 = 3580 \text{ LB}\cdot\text{IN}$

TORQUE IN SHAFT: $T_{A-B} = 0$; $T_{B-C} = 716 \text{ LB}\cdot\text{IN}$; $T_{C-D} = 2148 \text{ LB}\cdot\text{IN}$

$T_{D-F} = 1432 \text{ LB}\cdot\text{IN}$

FORCES ON SHAFT B: $F_1 - F_2 = T_B / r_B = 716/3 = 239 \text{ LB}$

$F_B = F_1 + F_2 = 1.5(F_1 - F_2) = 1.5(239) = 358 \text{ LB}$

$F_{Bx} = F_B \sin 30^\circ = 179 \text{ LB} \leftarrow$; $F_{By} = F_B \cos 30^\circ = 310 \text{ LB} \downarrow$

FORCES ON GEAR C: $W_{tc} = T_C / r_c = 1432/3 = 477 \text{ LB} \leftarrow = F_{cx}$

$W_{tc} = W_{tc} \tan 20^\circ = (477) \tan 20^\circ = 174 \text{ LB} \downarrow = F_{cy}$

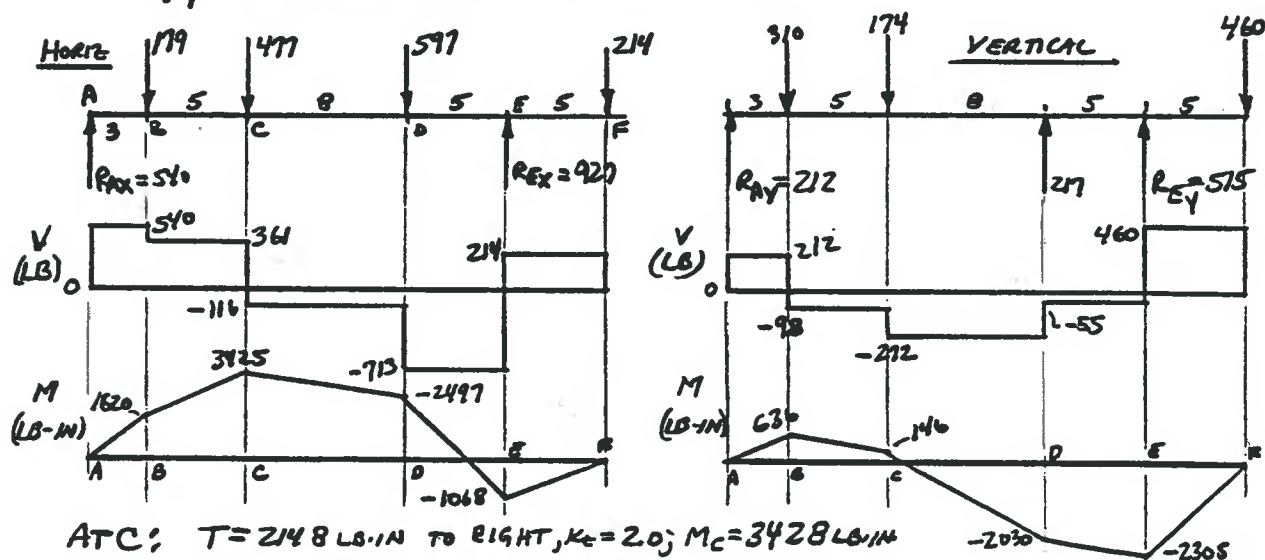
FORCES ON GEAR D: $W_{td} = T_D / r_d = 3580/6 = 597 \text{ LB} \leftarrow = F_{dx}$

$W_{td} = W_{td} \tan 20^\circ = (597) \tan 20^\circ = 217 \text{ LB} \uparrow = F_{dy}$

FORCES ON GEAR F: $W_{tf} = W_{tc} = 477 \text{ LB}$

$F_{fx} = 477 \sin 45^\circ - 174 \cos 45^\circ = 214 \text{ LB} \leftarrow$

$F_{fy} = 477 \cos 45^\circ + 174 \sin 45^\circ = 460 \text{ LB} \downarrow$



AT C: $T = 2148 \text{ LB}\cdot\text{IN}$ TO RIGHT, $K_t = 2.0$; $M_c = 3428 \text{ LB}\cdot\text{IN}$

$T = 716 \text{ LB}\cdot\text{IN}$ TO LEFT, $K_t = 3.0$

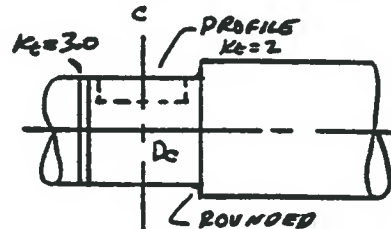
$D_{ca} = \left[\frac{32(3)}{\pi} \sqrt{\left(\frac{2.0(3428)}{15100} \right)^2 + \frac{3}{4} \left(\frac{2148}{57000} \right)^2} \right]^{1/3} = 2.40 \text{ IN}$

$D_{cl} = \left[\frac{32(3)}{\pi} \sqrt{\left(\frac{3.0(3428)}{15100} \right)^2 + \frac{3}{4} \left(\frac{716}{57000} \right)^2} \right]^{1/3} = 2.75 \text{ IN}$

INCREASE BY 6%: $D_c = 1.06(2.75)$

$D_c = 2.91 \text{ IN}$

SPECIFY $D_c = 3.00 \text{ IN}$ $C_s = 0.78$ OK



SAE 1020 CD: $S_y = 57 \text{ KSI}$; $S_u = 61 \text{ KSI}$
 $S_m = 22 \text{ KSI}$
 $S_m' = (0.85)(81)(1.2) = 15.1 \text{ KSI}$
 C_s CR

30

FIGURE P12-17

TORQUE AT FAN: $T_A = 63000(12)/475 = 1592 \text{ LB}\cdot\text{IN}$

" ON PULLEY D: $T_D = 63000(3.5)/475 = 464 \text{ LB}\cdot\text{IN}$

" ON SHEAVE C: $T_C = 63000(15.5)/475 = 2056 \text{ LB}\cdot\text{IN}$

TORQUE IN SHAFT: $T_{AC} = 1592 \text{ LB}\cdot\text{IN}$; $T_{CD} = 464 \text{ LB}\cdot\text{IN}$; $T_{DE} = 0$

FORCES ON A: $F_{Ax} = 0$; $F_{Ay} = 34 \text{ LB}$ THE FAN WOULD ALSO PRODUCE AN AXIAL THRUST FORCE BUT ITS EFFECT ON SHAFT DIA. IS SMALL.

FORCES ON C (V-BELT)
 $F_1 - F_2 = T_C / R_C = 2056 / 5.0 = 411 \text{ LB}$

$F_C = F_{Cx} = F_1 + F_2 = 1.5(F_1 - F_2) = 1.5(411) = 617 \text{ LB}$ \rightarrow ; $F_{Cy} = 0$

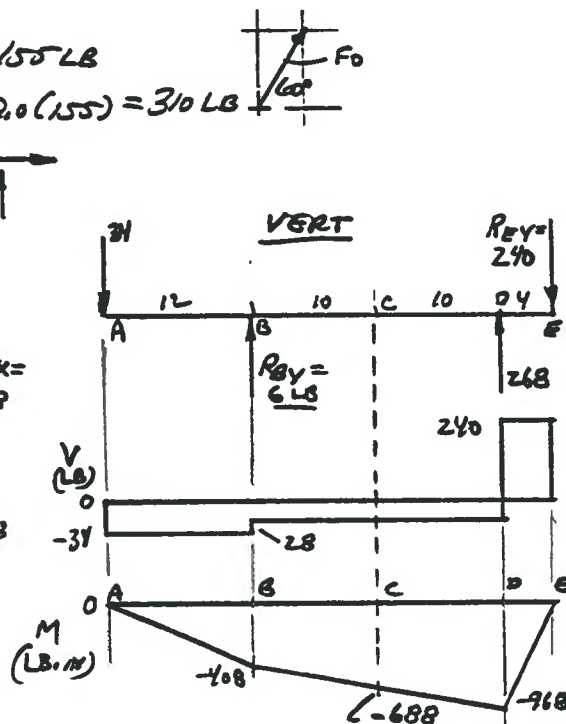
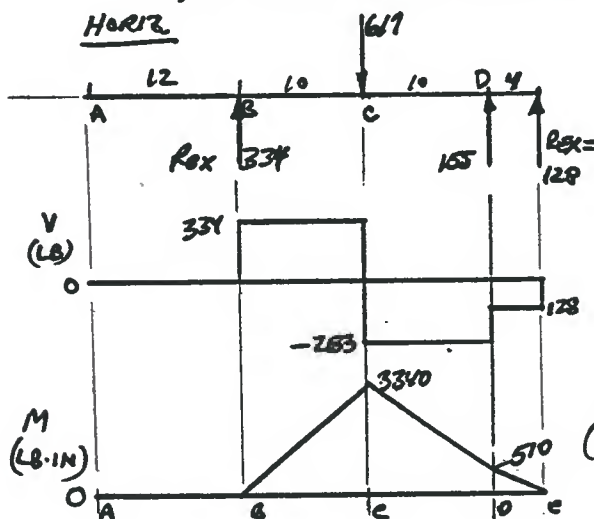
FORCES ON D: (FLAT BELT)

$F_1 - F_2 = T_D / R_D = 464 / 3 = 155 \text{ LB}$

$F_D = F_1 + F_2 = 2.0(F_1 - F_2) = 2.0(155) = 310 \text{ LB}$

$F_{Dx} = F_D \cos 60^\circ = 155 \text{ LB}$ \rightarrow

$F_{Dy} = F_D \sin 60^\circ = 268 \text{ LB}$ \uparrow



ATC: $T_{C \text{ LEFT}} = 1592 \text{ LB}\cdot\text{IN}$; $T_{C \text{ RIGHT}} = 464 \text{ LB}\cdot\text{IN}$

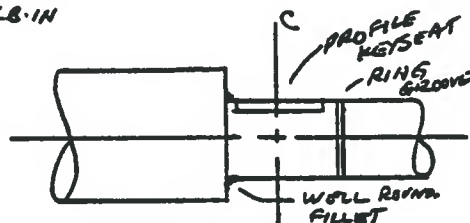
$M_C = \sqrt{3340^2 + 688^2} = 3410 \text{ LB}\cdot\text{IN}$

ATC & TO LEFT: $K_C = 2.0 \text{ KEYSEAT}$ $\frac{1}{3}$
 $D = \left[\frac{32(3)}{\pi} \sqrt{\frac{2.0(3410)^2}{.00} + \frac{3}{4} \left(\frac{1592}{90000} \right)^2} \right]^{\frac{1}{3}} = 2.00 \text{ IN}$

AT RIGHT OF C: $K_C = 3.0 - \text{RING GROOVE}$
 $D = \left[\frac{32(3)}{\pi} \sqrt{\frac{3.0(3410)^2}{.00} + \frac{3}{4} \left(\frac{464}{90000} \right)^2} \right]^{\frac{1}{3}} = 2.29 \text{ IN}$ CRITICAL

INCREASE BY 6%: $D = 1.06(2.29) = 2.42 \text{ IN. MIN}$

SPECIFY $D = 2.50 \text{ IN}$ $C_s = 0.79 \text{ OK}$



SAE 1144 CD: $S_{cy} = 90 \text{ KSI}$

$S_m = 100 \text{ KSI}$; $S_n = 38 \text{ KSI}$

$S'_m = (0.85)(0.8)(38) = 26.2 \text{ KSI}$
 $C_s \quad C_e$

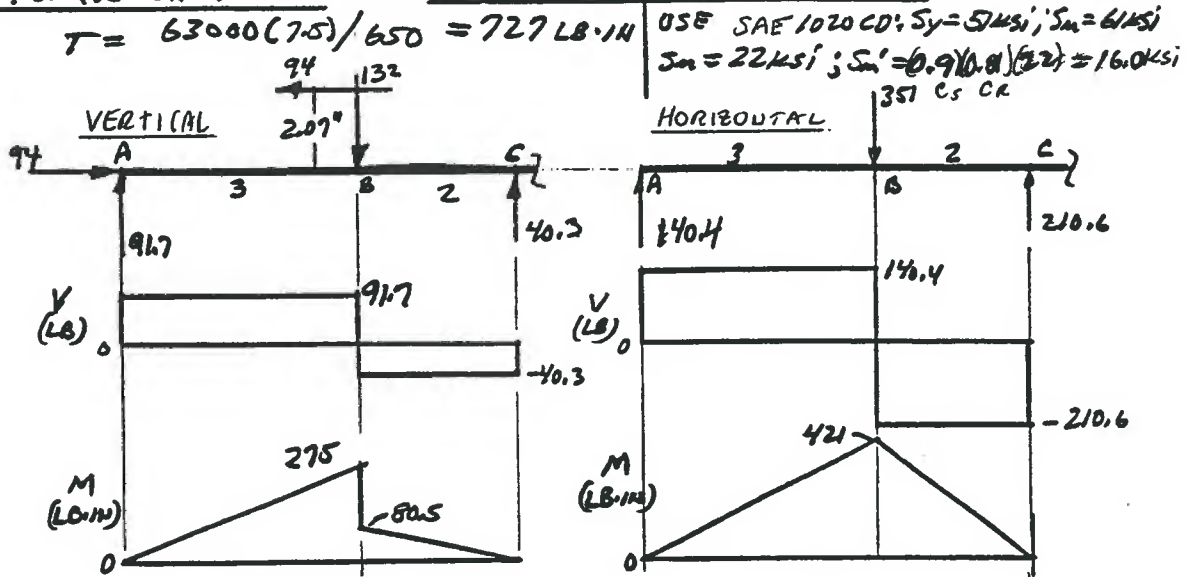
FIGURE P12-31

31

TORQUE ON GEAR B AND IN SHAFT FROM B TO COUPLING;

AND

32



AT B: $M = \sqrt{275^2 + 421^2} = 503 \text{ LB}\cdot\text{IN}$; USE $N=3$; $K_t=2.0$ (KEYSEAT)

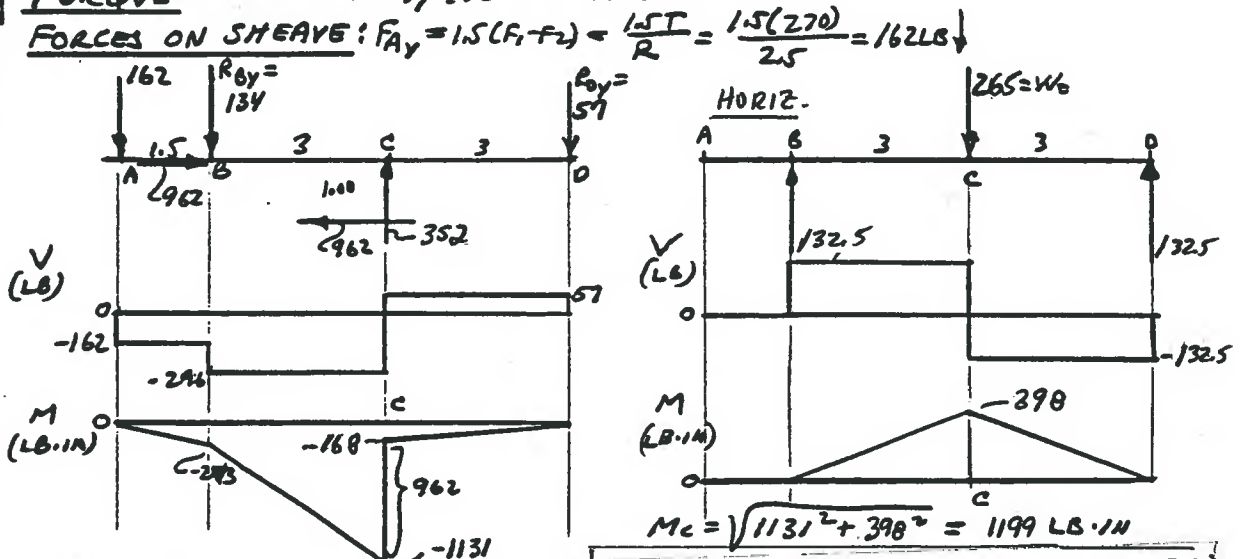
$D_B = \left[\frac{32(3)}{\pi} \sqrt{\left(\frac{2.0(503)}{16000} \right)^2 + \frac{3}{4} \left(\frac{727}{51000} \right)^2} \right]^{1/3} = 1.25 \text{ IN}$ $C_s > 0.9$ OK
 SPECIFY

33

TORQUE = $63000(7.5)/1750 = 270 \text{ LB}\cdot\text{IN}$

AND

34



ASSUME TORQUE IS NEGLIGIBLE, USE $N=4$

$\sigma = \frac{K_t M}{S} = \frac{1.5(1199)}{\pi(1.614)^3/32} = 4357 \text{ psi} = \frac{S_m'}{N}$

$S_m' = C_s C_R S_m = (0.83)(0.81) S_m = 0.67 S_m$

REQD $S_m' = N \sigma = 4(4357) = 17428 \text{ psi} = 0.67 S_m$

REQD $S_m = S_m' / 0.67 = 26000 \text{ psi}$

FROM FIG. 5-8; $S_m \approx 68 \text{ ksi}$

SPECIFY SAE 1040 CD; $S_u = 80 \text{ ksi}$; $S_y = 70 \text{ ksi}$

$S_m = 30 \text{ ksi}$; $S_m' = 0.67(30) = 20.1 \text{ ksi}$

CHECK WITH EQ. 12-24 WITH M AND T .

$D_c = \frac{32(4)}{\pi} \sqrt{\left(\frac{1.5(1199)}{20100} \right)^2 + \frac{3}{4} \left(\frac{270}{70000} \right)^2} = 1.54 \text{ IN}$ OK

THIS IS LESS THAN ACTUAL $D_c = 1.614 \text{ IN}$

FIGURE P12-35

35

TORQUES ON GEARS AND IN SHAFTS:

GEAR P, SHAFT 1: $T_1 = 63000(5)/1800 = 175 \text{ LB}\cdot\text{IN}$

GEARS B & C, SHAFT 2: $T_2 = 63000(5)/900 = 350 \text{ LB}\cdot\text{IN}$

GEAR Q, SHAFT 3: $T_3 = 63000(5)/300 = 1050 \text{ LB}\cdot\text{IN}$

SHAFT 2: (SEE CH. 10) $\phi_m = 14\frac{1}{2}^\circ$; $\psi = 45^\circ$

GEAR B

$$W_{tB} = T_2/r_B = 350/1.5 = 233 \text{ LB} \rightarrow \quad (\text{EQ 10-2})$$

$$W_{xB} = W_{tB} \tan \psi = 233 \tan 45^\circ = 233 \text{ LB} \quad (\text{EQ 10-8})$$

$$\phi_t = \tan^{-1} \left(\frac{\tan \phi_m}{\cos \psi} \right) = 20.1^\circ \text{ OR } \tan \phi_t = \frac{\tan \phi_m}{\cos \psi} = 0.366 \quad (\text{EQ 10-1})$$

$$W_{rB} = W_t \tan \phi_t = 233(0.366) = 85 \text{ LB} \quad (\text{EQ 10-7})$$

SIMILARLY:

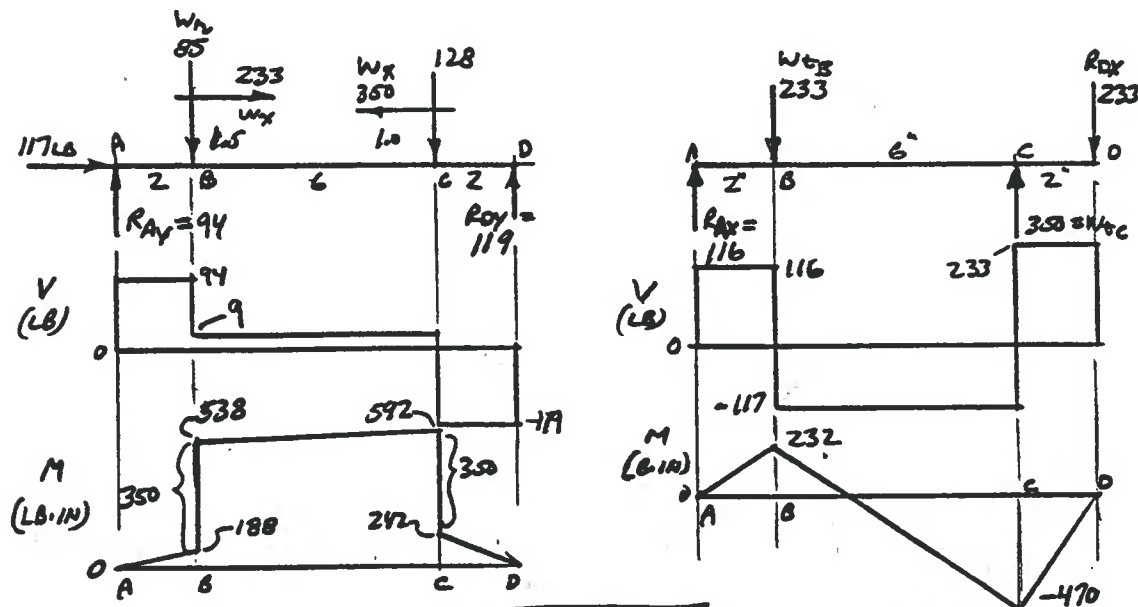
GEAR C

$$W_{tC} = T_2/r_C = 350/1 = 350 \text{ LB} \leftarrow$$

$$W_{xC} = W_{tC} \tan \psi = 350 \tan 45^\circ = 350 \text{ LB}$$

$$W_{rC} = W_t \tan \phi_t = 350(0.366) = 128 \text{ LB}$$

FOR SAE 4140 Q&T 1200; $S_y = 114 \text{ KSI}$; $S_u = 130 \text{ KSI}$; $S_m = 46 \text{ KSI}$ (FIG. 5-8)
 $S_m' = C_s C_R S_m = (0.9)(0.81)(46 \text{ KSI}) = 33.5 \text{ KSI}$



AT C: $T = 350 \text{ LB}\cdot\text{IN}$; $M_0 = \sqrt{592^2 + 470^2} = 756 \text{ LB}\cdot\text{IN}$; USE $K_t = 2.0$ (KEY)

$$D_c = \left[\frac{32(3)}{\pi} \sqrt{\left(\frac{2.0(756)}{33500} \right)^2 + \frac{3}{4} \left(\frac{350}{14000} \right)^2} \right]^{1/3} = 1.11 \text{ IN}$$

SPECIFY $D_c = 1.25 \text{ IN.}$; $C_s = 0.85$ OK

SHAFTS 1 AND 3 CAN BE DESIGNED SIMILAR TO PROBLEM 33.

DATA FROM FIGURE 10-8 to 10-12.

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PINION: $T = 263 \text{ LB}\cdot\text{IN}$; $M_E = \sqrt{220^2 + 95.8^2} = 240 \text{ LB}\cdot\text{IN}$; $N = 3$; $K_t = 2.0 (\text{KEY})$

SAE 1040 OQT 1200; $S_y = 63 \text{ KSI}$; $S_u = 93 \text{ KSI}$; $S_m = 36 \text{ KSI}$; $S'_m = (86)(0.8)(36) =$

$$D_E = \left[\frac{32(3)}{\pi} \sqrt{\left(\frac{2.0(240)}{25100} \right)^2 + \frac{3}{4} \left(\frac{263}{63000} \right)^2} \right]^{1/3} = 0.84 \text{ IN} \quad S'_m = 25.1 \text{ KSI}$$

GEAR: $T = 788 \text{ LB}\cdot\text{IN}$; $M_F = 390 \text{ LB}\cdot\text{IN}$; $N = 3$; $K_t = 2.0 (\text{KEY})$

$$D_F = \left[\frac{32(3)}{\pi} \sqrt{\left(\frac{2.0(390)}{25100} \right)^2 + \frac{3}{4} \left(\frac{788}{63000} \right)^2} \right]^{1/3} = 1.00 \text{ IN.} \quad C_s = 0.88 \text{ OK}$$

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$T_{\text{GEAR}} = 63000(4.0)/600 = 420 \text{ LB}\cdot\text{IN}$; $T_{SP} = 420/2 = 210 \text{ LB}\cdot\text{IN}$

$F_{t1} = F_{t2} = T_{SP}/5 = 210/5 = 42 \text{ LB}$

$\Gamma = \tan^{-1}(45/15) = 71.6^\circ$

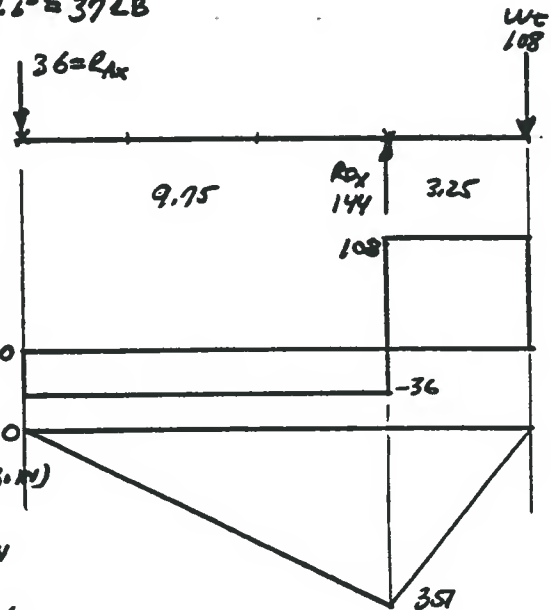
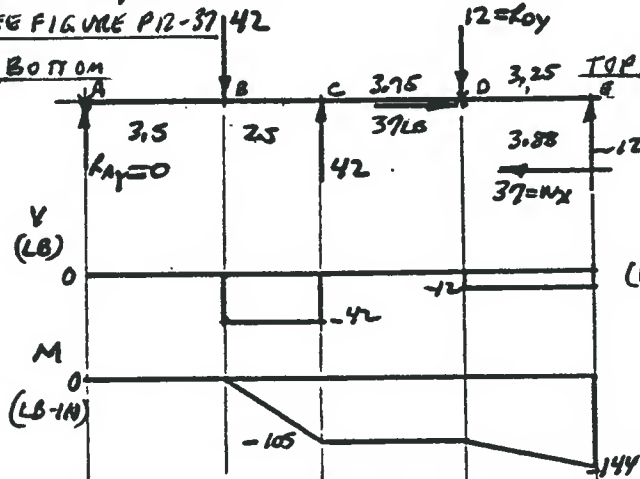
$W_{t2} = T_{SP}/R_m = 420/3.88 = 108 \text{ LB}$

$\gamma = \tan^{-1}(15/45) = 18.4^\circ$

$W_{x2} = W_t \tan \phi \cos \Gamma = (108) \tan 20^\circ \cos 71.6^\circ = 12 \text{ LB}$

$W_{x1} = W_t \tan \phi \sin \Gamma = (108) \tan 20^\circ \sin 71.6^\circ = 37 \text{ LB}$

SEE FIGURE P12-37



BEARING D: $M = \sqrt{105^2 + 35^2} = 366 \text{ LB}\cdot\text{IN}$

USE $N = 3$, $K_t = 2.5 (\text{FILLET})$

$$D_D = \left[\frac{32(3)}{\pi} \sqrt{\left(\frac{2.5(366)}{44600} \right)^2 + \frac{3}{4} \left(\frac{420}{152000} \right)^2} \right]^{1/3} = 0.86 \text{ IN}$$

SPECIFY $D = 1.00 \text{ IN}$ $C_s = 0.88$ OK

SAE 4140 OQT 1000

$S_y = 152 \text{ KSI}$; $S_u = 168 \text{ KSI}$

$S_m = 58 \text{ KSI}$; $S'_m = (95)(0.8)(58) =$

$S'_m = 44.6 \text{ KSI}$

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$P = 15.0 \text{ HP}$; $M_1 = 1725 \text{ RPM}$; $M_2 = 575 \text{ RPM}$; $M_3 = 287.5 \text{ RPM}$

TORQUE ON SHAFT 1 = $T_1 = T_A = 63000(15)/1725 = 548 \text{ LB}\cdot\text{IN.}$

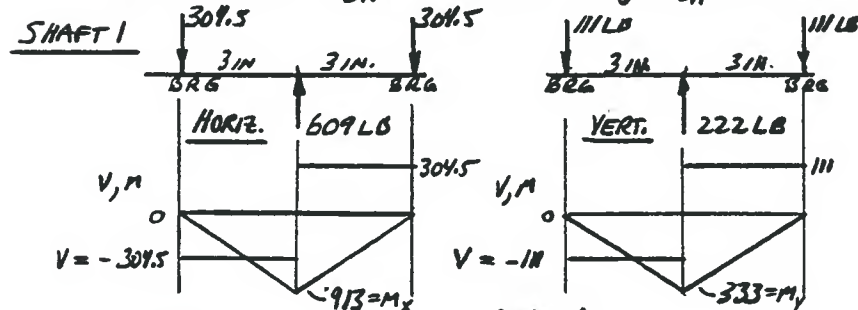
TORQUE ON SHAFT 2 = $T_2 = T_B = T_C = 63000(15)/575 = 1643 \text{ LB}\cdot\text{IN.}$

TORQUE ON SHAFT 3 = $T_3 = T_D = 63000(15)/287.5 = 3287 \text{ LB}\cdot\text{IN.}$

$W_{tB} = T_A/r_A = \frac{548 \text{ LB}\cdot\text{IN.}}{0.90 \text{ IN.}} = 609 \text{ LB} \leftarrow$

$W_{rB} = W_{tB} \tan \phi = 609 \text{ LB} \tan 20^\circ = 222 \text{ LB} \downarrow$

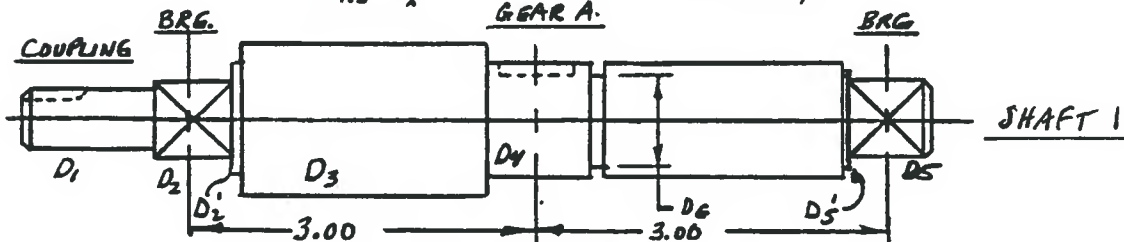
REACTIONS: $N_{tA} = 609 \text{ LB} \rightarrow$; $W_{rA} = 222 \text{ LB} \uparrow$



AT MIDDLE OF SHAFT 1

$$M = \sqrt{913^2 + 333^2} = 972 \text{ LB}\cdot\text{IN.}$$

$T = 548 \text{ LB}\cdot\text{IN.}$ FROM COUPLING TO GEAR A.



EQUATION (2-24) USED TO COMPUTE ALL DIAMETERS.

DIAMETER D_1 : $T = 548 \text{ LB}\cdot\text{IN.}$; $M = 0$; DESIGN FOR 0.999 RELIABILITY - $C_R = 0.75$

USE SAE 1040 CD STEEL: $S_y = 71,000 \text{ PSI}$; $S_u = 80,000 \text{ PSI}$; 12% ELONGATION
 $S_m = 30,000 \text{ PSI}$; $S_m' = S_m C_S C_R = (30,000)(0.9)(0.75) = 20,250 \text{ PSI}$; LET $N = 3$.

THEN $D_{1 \min} = 0.589 \text{ IN.}$

DIAMETER D_2 : SAME CONDITIONS AS D_1 ; $D_{2 \min} = 0.589 \text{ IN.}$

DIAMETER D_3 : DEPENDS ON D_4 .

DIAMETER D_4 : $M = 972 \text{ LB}\cdot\text{IN.}$; $T = 548 \text{ LB}\cdot\text{IN.}$ AT SHOULDER AND KEYSEAT.
 $(K_t \approx 2.5)$ $(K_r \approx 2.0)$

AT SHOULDER: $D_{4 \min} = 1.54 \text{ IN.}$

AT RING GROOVE: $T = 0$, $K_t \approx 3.0$; $D_{6 \min} = 1.64 \text{ IN.}$

INCREASE BY 6% FOR D_4 : $D_4 \approx 1.06(1.64) = 1.74 \text{ IN.}$ GOVERNING VALUE

DIAMETER D_5 : $M = 0$; $T = 0$: VERY SMALL DIA. REQD TO RESIST SHEAR.

BEARING SEATS D_2 , D_5 : ASSUME BEARINGS WITH BORE = 0.7874 IN. (20mm)

CAN BE FOUND TO CARRY RADIAL LOADS. CHAPTER 14, TABLE 14-3

SPECIFICATIONS:

BEARING NO. 6204

$D_1 = 0.750 \text{ IN.}$

D_6 SPECIFIED BY RETAINING RING MFR.

$D_2 = D_5 = 0.7874 \text{ IN.}$

$D_3 = 2.00 \text{ IN.}$

$D_4 = 1.80 \text{ IN.}$

RELIEF PROVIDED ON LEFT SIDE OF D_3 AND RIGHT END OF D_4
 TO ENSURE THAT OUTER RACE OF BEARING 6 DOES NOT CONTACT
 ROTATING SHAFT. $D_2' = D_4' = 0.969 \text{ IN.}$ (SHAFT SHOULDER)

C_S CHECKED FOR ALL DIAMETERS - OK

(CONTINUED)

38

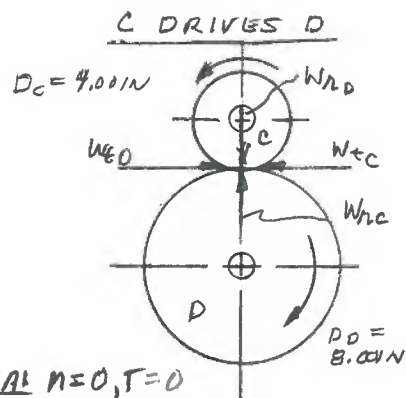
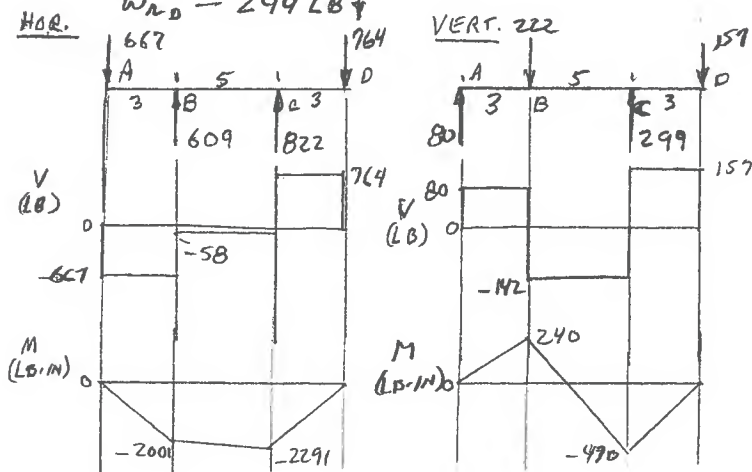
(CONTINUED) SHAFT 2 $T_2 = 1643 \text{ LB}\cdot\text{IN}$

$$W_{tc} = \frac{T_2}{R_c} = \frac{1643 \text{ LB}\cdot\text{IN}}{2.00 \text{ IN}} = 822 \text{ LB} \rightarrow$$

$$W_{td} = 822 \text{ LB} \rightarrow$$

$$W_{rc} = W_{tc} \tan \phi = 822 \tan 20^\circ = 299 \text{ LB} \uparrow$$

$$W_{rd} = 299 \text{ LB} \downarrow$$

AT A: $M=0, T=0$

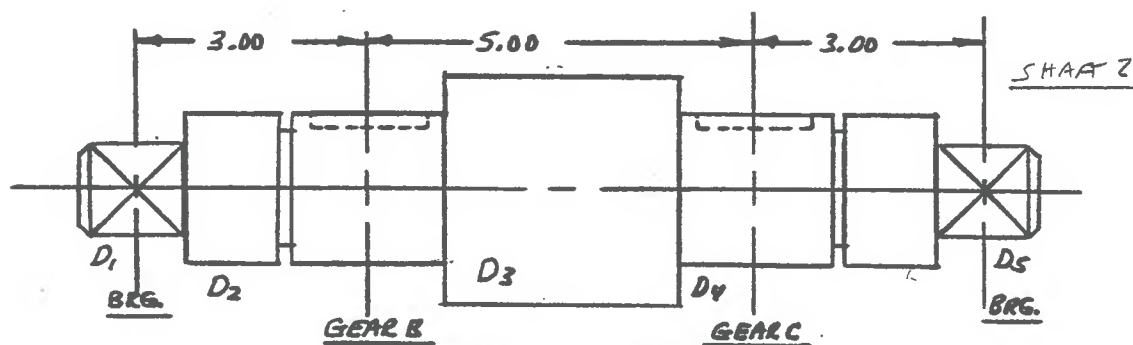
$$V_A = \sqrt{667^2 + 80^2} = 671 \text{ LB}$$

$$B: M_B = \sqrt{2001^2 + 240^2} = 2015 \text{ LB}\cdot\text{IN}$$

$$C: M_C = \sqrt{2291^2 + 470^2} = 2339 \text{ LB}\cdot\text{IN}$$

$$D: M=0, T=0$$

$$V_D = \sqrt{764^2 + 157^2} = 780 \text{ LB}$$

SAE 1040 CD STEEL: $S_y = 71000 \text{ PSI}$; $S_u = 80000 \text{ PSI}$; 12% EL. $S_m = 30000 \text{ PSI}$; $S_m' = 20250 \text{ PSI}$ [SAME AS SHAFT 1]D2: $M_B = 2015 \text{ LB}\cdot\text{IN}$; $T = 1643 \text{ LB}\cdot\text{IN}$ AT KEYSEAT + SHOULDER ($K_t = 2.5$)

$$D_{2 \text{ MIN}} = 1.968 \text{ IN. AT SHOULDER} \quad (K_t = 2.0)$$

AT RING GROOVE: $D_{2 \text{ MIN}} = (1.06)(2.089) = 2.215 \text{ IN. GOVERNS}$ D4: $M_C = 2339 \text{ LB}\cdot\text{IN}$; $T = 1643 \text{ LB}\cdot\text{IN}$ AT KEYSEAT AND SHOULDER ($K_t = 2.5$)

$$D_{4 \text{ MIN}} = 2.068 \text{ IN}$$

AT RING GROOVE: $T=0$, $K_t = 3.0$; $M = 2339 \text{ LB}\cdot\text{IN}$

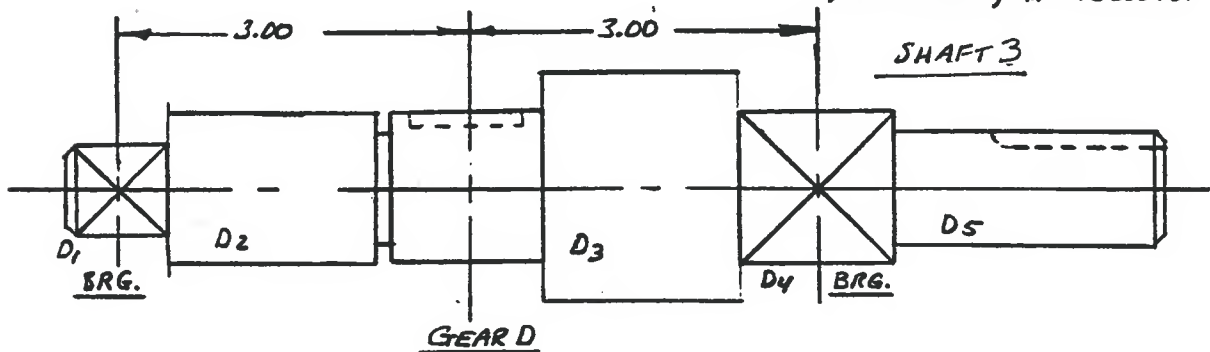
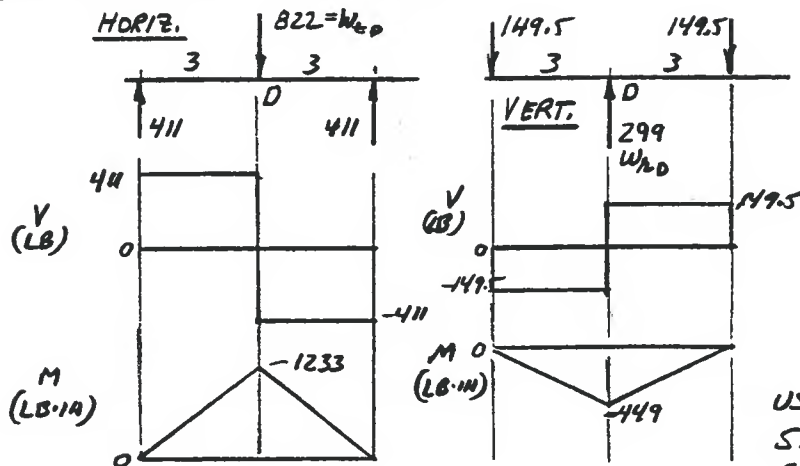
$$D_{4 \text{ MIN}} = (1.06)(2.1959) = 2.328 \text{ IN. GOVERNS}$$

D5: RIGHT BRG.: $M=0$; $T=0$; $V_D = 780 \text{ LB}$; $K_t = 2.5$

$$D_{5 \text{ MIN}} = \sqrt{2.94 K_t V_N / S_m'} = \sqrt{2.94 (2.5) (780) (3) / 20250} = 0.922 \text{ IN.}$$

D1: $V_A = 671 \text{ LB} \rightarrow D_{1 \text{ MIN}} = 0.855 \text{ IN.}$ LET: $D_2 = 2.250 \text{ IN}$; $D_4 = 2.400 \text{ IN}$; $D_3 = 2.600 \text{ IN}$ D1 AND D5 DEPEND ON BEARING SELECTION.

38 (CONTINUED) SHAFT 3



DIAMETER D₂: $M = 1312 \text{ LB}\cdot\text{IN}$; $T = 3287 \text{ LB}\cdot\text{IN}$, $K_t = 2.5$ AT SHOULDER.

$D_2 = 1.79 \text{ IN.}$ AT SHOULDER.

AT GROOVE: $D_6 = 1.88 \text{ IN}$ FOR $K_t = 3.0$; $T = 0$.

INCREASE BY 6%: $D_2 \approx 1.06(1.88) = 2.00 \text{ IN. GOVERNS}$

DIAMETER D_{4 MIN}; D_{5 MIN}: $M = 0$; $T = 3287 \text{ LB}\cdot\text{IN}$; $D_{4 \text{ MIN}} = 1.07 \text{ IN.}$

SPECIFY:

$D_1 = 1.3780 \text{ IN (35 mm)}$, BRG. 6207

$D_2 = 2.000 \text{ IN}$

$D_3 = 2.250 \text{ IN.}$

$D_4 = 1.3780 \text{ IN (35 mm)}$ BRG. 6207

$D_5 = 1.25 \text{ IN}$

[CHECKED C_s OK]

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$P = 12.0 \text{ HP}$, $n_{\text{motor}} = 1150 \text{ RPM}$, FIGURE P12-39

MOTOR SHAFT:

$$\text{NET DRIVING FORCE} = F_H = T/D/2$$

$$T = 63000(\text{HP})/n = 63000(12)/1150 = 657 \text{ LB}\cdot\text{IN.}$$

$$F_H = 657 \text{ LB}\cdot\text{IN.} / (5.6 \text{ IN.} / 2) = 235 \text{ LB.}$$

$$\text{BENDING FORCE} = F_B = 1.5 F_H = 352 \text{ LB.}$$

$$F_{Bx} = F_B \sin 35^\circ = (352 \text{ LB.})(\sin 35^\circ) = 202 \text{ LB.}$$

$$F_{By} = F_B \cos 35^\circ = (352 \text{ LB.})(\cos 35^\circ) = 288 \text{ LB.}$$

FORCES ACT ON MOTOR SHAFT. DIRECTIONS AS VIEWED FROM BEHIND THE LEFT END OF THE MOTOR.

REDUCER INPUT SHAFT: FORCES SHOWN AS VIEWED FROM RIGHT END FOR CONSISTENCY WITH VIEWS OF GEAR SYSTEM IN FIGURE P12-38.

$$\text{SHAFT SPEED: } n_1 = n_{\text{motor}} \left(\frac{D_1}{D_2} \right) = 1150 \text{ RPM} \left(\frac{5.6}{8.4} \right) = 767 \text{ RPM}$$

$$T_1 = 63000(12)/767 = 986 \text{ LB}\cdot\text{IN.}$$

$$\text{FORCES AT SHEAVE: } F_H = T_1/(D_1/2) = (986 \text{ LB}\cdot\text{IN.})/(8.4 \text{ IN.}/2) = 235 \text{ LB.}$$

$$F_B = 1.5 F_H = 352 \text{ LB.}; F_{Bx} = 202 \text{ LB.}; F_{By} = 288 \text{ LB.}$$

(SEE MOTOR SHAFT ANALYSIS)

FROM PROBLEM 38:

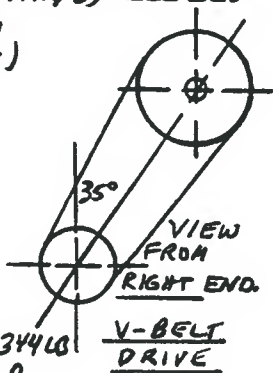
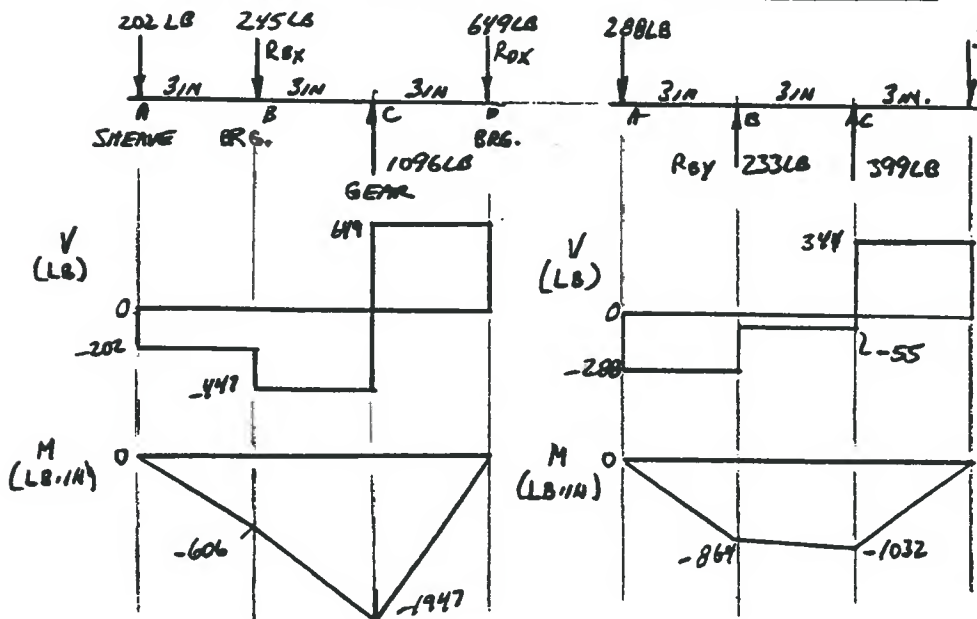
$$\text{GEAR A: } W_{GA} = T_1/n_A = 986 \text{ LB}\cdot\text{IN.}/0.9 \text{ IN.} = 1096 \text{ LB.}$$

$$W_{GA} = W_{CA} \tan \phi = (1096 \text{ LB.}) \tan 20^\circ = 399 \text{ LB.}$$

POSITION OF SHEAVE ON SHAFT 1 ASSUMED

HORIZONTAL PLANE

VERTICAL PLANE



RESULTANT MOMENTS:

$$M_B = \sqrt{606^2 + 864^2}$$

$$M_B = 1055 \text{ LB}\cdot\text{IN.}$$

$$M_C = \sqrt{1947^2 + 1032^2}$$

$$M_C = 2204 \text{ LB}\cdot\text{IN.}$$

BEARING FORCES:

$$R_B = \sqrt{245^2 + 233^2}$$

$$R_B = 338 \text{ LB.}$$

$$R_D = \sqrt{649^2 + 344^2}$$

$$R_D = 735 \text{ LB.}$$

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CONTINUED

DESIGN OF SHAFT 1 WOULD BE COMPLETED IN A MANNER SIMILAR TO THAT SHOWN IN PROBLEM 38.

SHAFT 2: SEE PROBLEM 38 FOR ANALYSIS AND DESIGN PROC.

FORCES: $W_{tB} = W_{cA} = 1086 \text{ LB} \leftarrow$; $W_{rB} = W_{rA} = 399 \text{ LB} \uparrow$ GEAR B

SHAFT 2 SPEED = $M_2 = M_1 \frac{N_A}{N_B} = 767 \times \frac{18}{54} = 256 \text{ RPM}$

$T_2 = 63000 (12) / 256 = 2958 \text{ LB}\cdot\text{IN}$

$W_{tC} = T_2 / r_C = 2958 \text{ LB}\cdot\text{IN} / 2.00 \text{ IN} = 1479 \text{ LB} \rightarrow$ GEAR C

$W_{rC} = W_{tC} \tan 20^\circ = 538 \text{ LB} \downarrow$

SHAFT 3: SEE PROBLEM 38 FOR ANALYSIS AND DESIGN PROCEDURE.

FORCES: $W_{tD} = W_{tC} = 1479 \text{ LB} \leftarrow$; $W_{rD} = W_{rC} = 538 \text{ LB} \uparrow$ GEAR D

CHAIN SPROCKET AT END OF SHAFT 3:

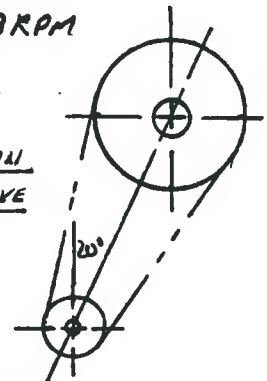
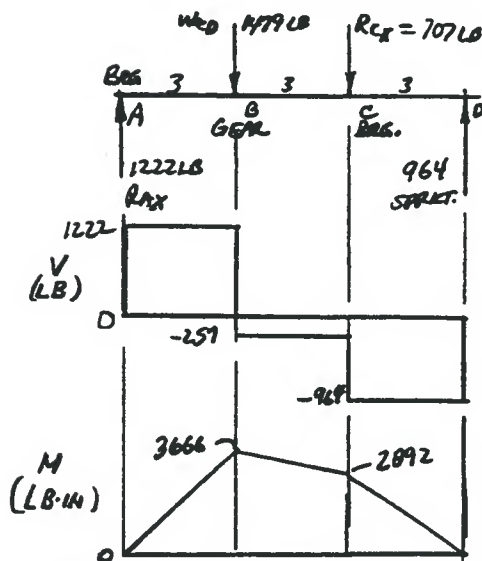
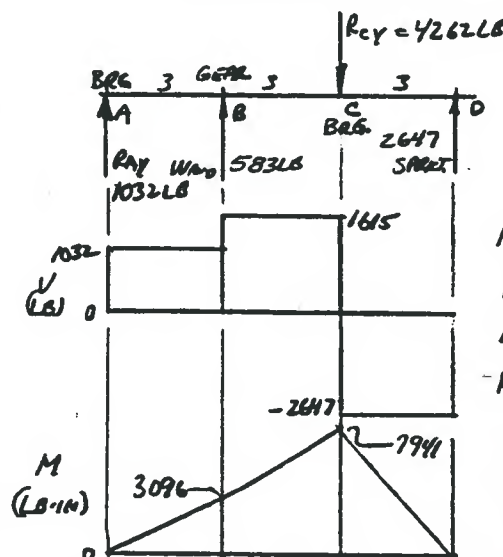
SPEED: $M_3 = M_2 \cdot \frac{N_2}{N_3} = 256 \text{ RPM} \cdot \frac{24}{48} = 128 \text{ RPM}$

$T_3 = 63000 (12) / 128 = 5917 \text{ LB}\cdot\text{IN}$

$F_N = F_B = \frac{T_3}{R} = \frac{5917 \text{ LB}\cdot\text{IN}}{(4.2 \text{ IN} / 2)} = 2817 \text{ LB} \uparrow$

$F_{Bx} = F_B \sin 20^\circ = (2817 \text{ LB}) \sin 20^\circ = 964 \text{ LB} \rightarrow$ CHAIN DRIVE

$F_{By} = F_B \cos 20^\circ = (2817 \text{ LB}) \cos 20^\circ = 2647 \text{ LB} \uparrow$

HORIZONTAL PLANEVERTICAL PLANERESULTS:

$$M_B = \sqrt{3666^2 + 3096^2}$$

$$M_B = 4798 \text{ LB}\cdot\text{IN}$$

$$M_C = \sqrt{2892^2 + 7941^2}$$

$$M_C = 8451 \text{ LB}\cdot\text{IN}$$

BEARING FORCES:

$$R_A = \sqrt{1222^2 + 1032^2}$$

$$R_A = 1600 \text{ LB}$$

$$R_C = \sqrt{707^2 + 4262^2}$$

$$R_C = 4320 \text{ LB}$$

CONVEYOR SHAFT FORCES: $F_{Bx} = 964 \text{ LB} \leftarrow$; $F_{By} = 2647 \text{ LB} \uparrow$

SPEED: $M_C = M_3 \cdot \frac{42}{10.6} = 128 \cdot \frac{42}{10.6} = 50.6 \text{ RPM}$; $T_C = 63000 (12) / 50.6 = 14,932 \text{ LB}\cdot\text{IN}$

FIGURE P12-40: SHAFT 2: $M = 480 \text{ RPM}$: POWER IN AT C = 22.5 kW .

POWER OUT AT A = 15 kW : POWER OUT AT E = 7.5 kW .

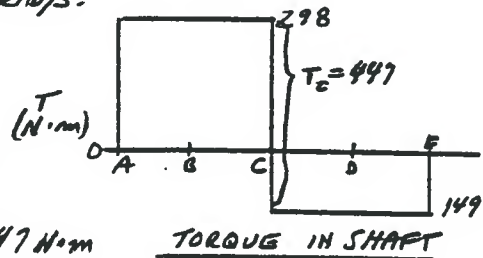
COAL CRUSHER - USE $N = 4$ BECAUSE OF IMPACT AND SHOCK.

$$n = 480 \frac{\text{REV}}{\text{MIN.}} \times \frac{2\pi \text{ RAD}}{\text{REV.}} \times \frac{1 \text{ MIN.}}{60 \text{ SEC.}} = 50.27 \text{ RAD/S.}$$

$$T_{AC} = \frac{P}{n} = \frac{15 \times 10^3 \text{ N}\cdot\text{m/s}}{50.27 \text{ RAD/S}} = 298 \text{ N}\cdot\text{m}$$

$$T_{CE} = \frac{P}{n} = \frac{7.5 \times 10^3 \text{ N}\cdot\text{m/s}}{50.27 \text{ RAD/S}} = 149 \text{ N}\cdot\text{m}$$

$$\text{TORQUE ON GEAR C} = \frac{22.5 \times 10^3 \text{ N}\cdot\text{m/s}}{50.27 \text{ RAD/S}} = 447 \text{ N}\cdot\text{m}$$



FORCES:

$$\text{GEAR A: } W_{tA} = \frac{T_A}{r_A} = \frac{298 \text{ N}\cdot\text{m} \times 10^3 \text{ mm}}{50 \text{ mm}} = 5960 \text{ N} \leftarrow$$

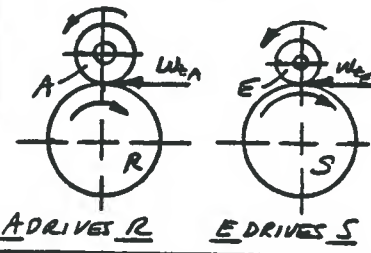
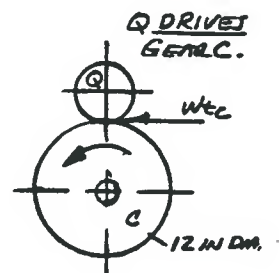
$$W_{tA} = W_{tC} \tan 20^\circ = (5960 \text{ N}) \tan 20^\circ = 2169 \text{ N} \uparrow$$

$$\text{GEAR C: } W_{tC} = \frac{T_C}{r_C} = \frac{447 \text{ N}\cdot\text{m} \times 10^3}{150 \text{ mm}} = 2980 \text{ N} \leftarrow$$

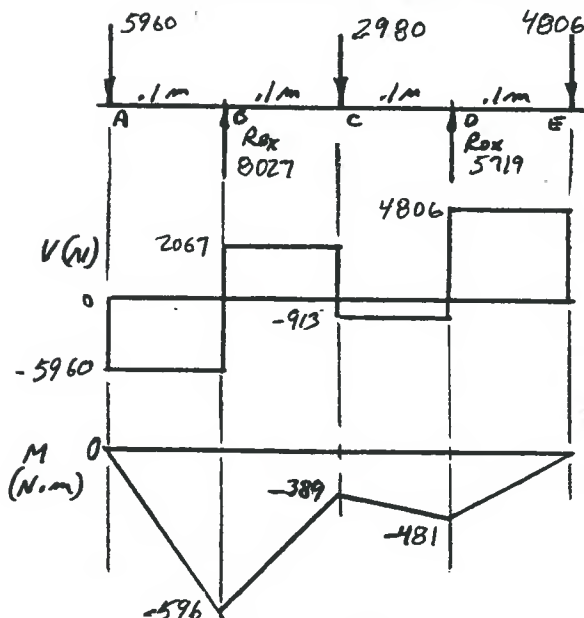
$$W_{tC} = W_{tE} \tan 20^\circ = (2980 \text{ N}) \tan 20^\circ = 1065 \text{ N} \uparrow$$

$$\text{GEAR E: } W_{tE} = \frac{T_E}{r_E} = \frac{149 \text{ N}\cdot\text{m} \times 10^3}{31 \text{ mm}} = 4806 \text{ N} \leftarrow$$

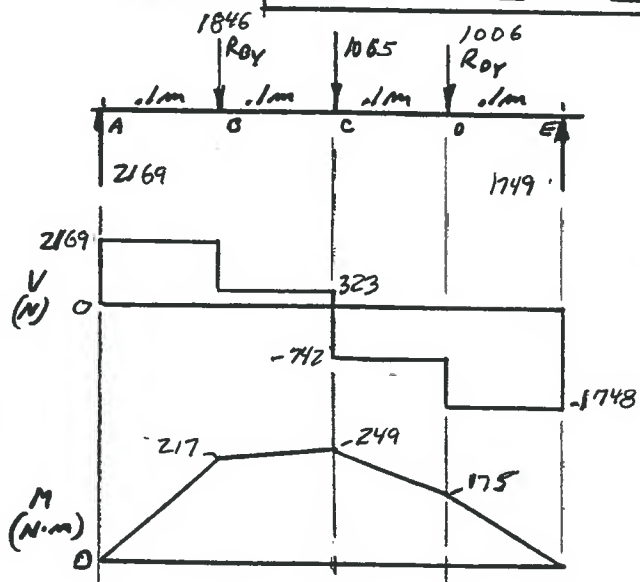
$$W_{tE} = W_{tA} \tan 20^\circ = (4806 \text{ N}) \tan 20^\circ = 1749 \text{ N} \uparrow$$



HORIZONTAL PLANE



VERTICAL

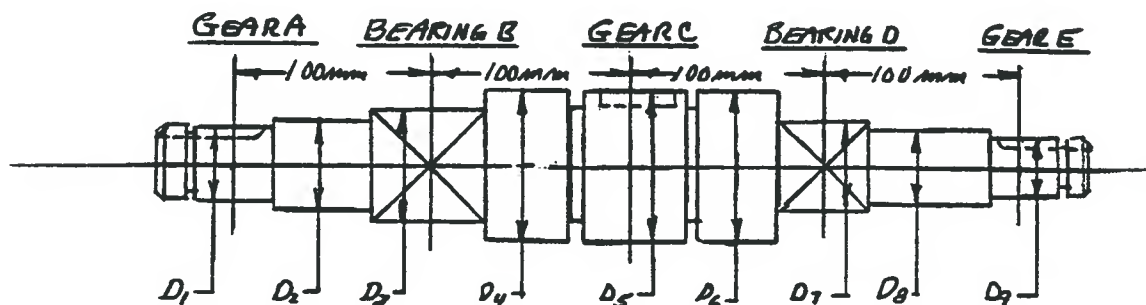


$$M_B = [596^2 + 217^2]^{1/2} = 634 \text{ N}\cdot\text{m}; \quad M_C = [389^2 + 249^2]^{1/2} = 462 \text{ N}\cdot\text{m}; \quad M_D = [481^2 + 175^2]^{1/2} = 512 \text{ N}\cdot\text{m}$$

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CONTINUED

BEARING FORCES: $R_B = \sqrt{8027^2 + 1846^2} = 8237 \text{ N}$; $R_D = \sqrt{5719^2 + 1006^2} = 5807 \text{ N}$

PROPOSED SHAFT DESIGN:

ASSUME ALL FILLETS ARE SMALL RADII WITH $K_t \approx 2.5$, EXCEPT r_2, r_6
 USE $K_t = 3.0$ AT RING GROOVES WITH SHAFT DIA. $\approx 1.06 \times$ GROOVE DIA.
 USE $K_t = 1.5$ AT r_2 AND r_6 (WELL ROUNDED)

MATERIAL SELECTION: SAE 4140 OQT 1000, $S_u = 1160 \text{ MPa}$, $S_y = 1050 \text{ MPa}$

17% ELONGATION - GOOD STRENGTH AND DUCTILITY.

FROM FIG. 5-9: $S_m = 400 \text{ MPa}$.

SELECT $C_s = 0.80$ (FWD $\approx 65 \text{ mm}$ OR LESS); $C_R = 0.81$ (0.99 RELIABILITY)

$S_m' = C_s C_R S_m = (0.80)(0.81)(400 \text{ MPa}) = 259 \text{ MPa} = 259 \text{ N/mm}^2$

SOLUTION FOR DIAMETERS USING EQ. 9-22 - SUMMARY: ($N=4$)

LOCATION	K_t	$M (\text{N}\cdot\text{mm})$	$T (\text{N}\cdot\text{mm})$	D_{MIN}	SPECIFIED D
D_1	2.5	0	298,000	21.56	50.0 mm
D_2	1.5	634,000	298,000	53.13	60.0 mm
D_3	2.5	634,000	298,000	62.96	65.0 mm
D_4	3.0	525,000*	298,000	62.82 x 1.06 = 66.6	80.0 mm
D_5	2.5	462,000	298,000	58.67	
D_6	3.0	462,000	149,000	60.19 x 1.06 = 63.8	
D_7	2.5	512,000	149,000	58.62	60.0 mm
D_8	1.5	512,000	149,000	49.45	55.0 mm
D_9	2.5	0	149,000	17.11	45.0 mm

NOTES: D_3 AND D_7 ARE STANDARD BEARING BORES FROM TABLE 14-3.

* MOMENT AT D_4 ESTIMATED BETWEEN POINTS B AND C.

80.0 mm USED FOR D_4, D_5 , AND D_6 TO PROVIDE SHOULDER FOR BEARINGS AT B AND D.

D_2 AND D_8 PROVIDE EASE OF INSTALLATION FOR BEARINGS.

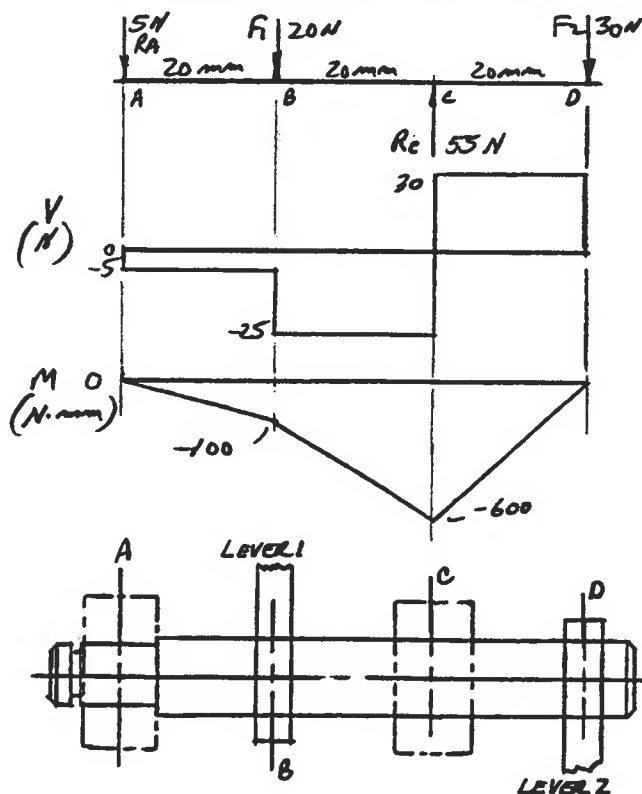
D_1 AND D_9 MADE LARGER THAN REQUIRED FOR COMPATIBILITY WITH ADJACENT DIAMETERS AND TO WITHSTAND MOMENTS AT SHOULDERS.

FINAL STRESSES MUST BE CHECKED AFTER SPECIFYING FILLET RADII, GROOVE GEOMETRY, AND BEARING SELECTION.

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FIGURE P12-41 DESIGN SHAFT AND LEVERS. $F_1 = 20\text{ N}$

$$F_2 = F_1(60\text{ mm})/40\text{ mm} = 30\text{ N} \quad \text{TORQUE} = F_1 \cdot 60 = 1200\text{ N}\cdot\text{mm} \text{ FROM B TO D.}$$



USE SAE 1137 CD STEEL

$$S_u = 676\text{ MPa}; S_y = 565\text{ MPa}$$

$$S_m = 260\text{ MPa (FIG. 5-8)}$$

$$\text{LET } C_s = 0.9, C_r = 0.75$$

$$S_m' = (0.9)(0.75)(260) = 175\text{ MPa}$$

WIPER MECHANISM HAS AN OSCILLATING MOTION. BOTH BENDING AND TORSION WILL BE VARYING.

SHAFT RESTRAINED IN BEARING AT A BUT CAN FLOAT IN BEARING C.

ASSUME $K_t = 2.5$ AT PILLT TO RIGHT OF A. ASSUME $K_t = 1.0$ AT C.

ASSUME LEVERS ARE INSTALLED WITH A LIGHT PRESS FIT AND TACK WELDED IN POSITION. USE $K_t = 3.0$ FOR WELD AREA.

DESIGN EQUATION 12-24 MUST BE MODIFIED FOR VARYING TORQUE.

ADD K_t FOR TORSION. SUBSTITUTE S_m' FOR S_y . THEN:

$$D = \left[\frac{32N}{\pi} \sqrt{\left(\frac{K_{tB} M}{S_m'} \right)^2 + \frac{3}{4} \left(\frac{K_{tT} T}{S_m'} \right)^2} \right]^{1/3}$$

$$\text{AT C:} \quad D = \left[\frac{32(3)}{\pi} \sqrt{\left(\frac{1.0(600)}{175} \right)^2 + \frac{3}{4} \left(\frac{1.0(1200)}{175} \right)^2} \right]^{1/3} = 5.94\text{ mm}$$

$$\text{AT B:} \quad D = \left[\frac{32(3)}{\pi} \sqrt{\left(\frac{3.0(100)}{175} \right)^2 + \frac{3}{4} \left(\frac{3.0(1200)}{175} \right)^2} \right]^{1/3} = 8.18\text{ mm}$$

USE $D \geq 10.0\text{ mm}$
AT B, C, D.
USE $D \geq 8.0\text{ mm}$
AT A.

LEVER DESIGN: USE FLAT STOCK, 4.0 mm THICK $b = 6$

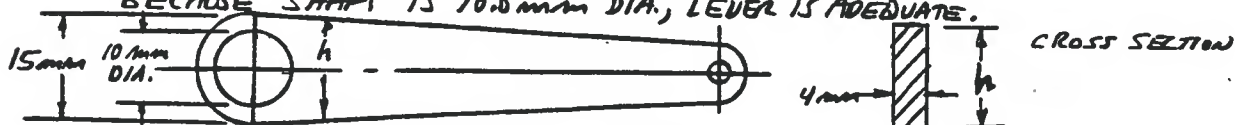
$$\text{SAME MATERIAL AS SHAFT: } S_m' = 175\text{ MPa}; \sigma_b = \frac{S_m'}{N} = \frac{175}{3} = 58.3\text{ MPa}$$

$$M_{\max} = (20\text{ N})(60\text{ mm}) = 1200\text{ N}\cdot\text{mm} \quad \sigma = M/S$$

$$\text{REQ'D } S = \frac{M}{\sigma_b} = \frac{1200\text{ N}\cdot\text{mm}}{58.3\text{ N/mm}} = 20.57\text{ mm}^3 = bh^2/6$$

$$\text{REQ'D } h = \sqrt{6S/b} = \sqrt{6(20.57\text{ mm}^3)/4.0\text{ mm}} = 5.55\text{ mm AT SHAFT.}$$

BECAUSE SHAFT IS 10.0 mm DIA., LEVER IS ADEQUATE.



CHAPTER 13 TOLERANCES AND FITS

1. LOOSE - RCB: HOLE $\begin{smallmatrix} +5 \\ -0 \end{smallmatrix}$; SHAFT $\begin{smallmatrix} -7 \\ -10.5 \end{smallmatrix}$; CLEARANCE $\begin{smallmatrix} +7 \\ +15.5 \end{smallmatrix}$
 LIMITS: HOLE $\frac{3.5050}{3.5000}$ SHAFT $\frac{3.4930}{3.4895}$ CL 0.0070 to 0.0155

2. PRECISION - RC2: HOLE $\begin{smallmatrix} +0.4 \\ -0 \end{smallmatrix}$; SHAFT $\begin{smallmatrix} -0.25 \\ -0.55 \end{smallmatrix}$; CL $\begin{smallmatrix} +0.25 \\ +0.95 \end{smallmatrix}$
 LIMITS: HOLE $\frac{0.50040}{0.50000}$ SHAFT $\frac{0.49975}{0.49945}$ CL 0.00025 to 0.00095

3. LOOSE - RCB: HOLE $\begin{smallmatrix} +2.8 \\ -0 \end{smallmatrix}$; SHAFT $\begin{smallmatrix} -3.5 \\ -5.1 \end{smallmatrix}$; CL $\begin{smallmatrix} +3.5 \\ +7.9 \end{smallmatrix}$
 ADJUST TOLERANCES FOR BASIC SHAFT SYSTEM - ADD 3.5.

HOLE $\begin{smallmatrix} +6.3 \\ +3.5 \end{smallmatrix}$; SHAFT $\begin{smallmatrix} 0 \\ -1.6 \end{smallmatrix}$; CL $\begin{smallmatrix} +3.5 \\ +7.9 \end{smallmatrix}$
 LIMITS: HOLE $\frac{0.6313}{0.6285}$ SHAFT $\frac{0.6250}{0.6234}$ CL 0.0035 to 0.0079

4. CLOSE FIT - RELIABLE MOTION - RC5
 HOLE $\begin{smallmatrix} +1.2 \\ -0 \end{smallmatrix}$; SHAFT $\begin{smallmatrix} -1.6 \\ -2.4 \end{smallmatrix}$; CL $\begin{smallmatrix} +1.6 \\ +3.6 \end{smallmatrix}$
 LIMITS: HOLE $\frac{0.8012}{0.8000}$ SHAFT $\frac{0.7984}{0.7976}$ CL 0.0016 to 0.0036

5. LOOSE - RCB: HOLE $\begin{smallmatrix} +4.0 \\ -0 \end{smallmatrix}$; PIN $\begin{smallmatrix} -5.0 \\ -7.5 \end{smallmatrix}$; CL $\begin{smallmatrix} +5.0 \\ +11.5 \end{smallmatrix}$
 LIMITS: HOLE $\frac{1.2540}{1.2500}$ PIN $\frac{1.2450}{1.2425}$ CL 0.0050 to 0.0115

6. LOOSE - RCB: HOLE $\begin{smallmatrix} +5.0 \\ -0 \end{smallmatrix}$; PIN $\begin{smallmatrix} -7.0 \\ -10.5 \end{smallmatrix}$; CL $\begin{smallmatrix} +7.0 \\ +15.5 \end{smallmatrix}$
 ADJUST TOLERANCES FOR BASIC SHAFT SYSTEM - ADD 7.0

HOLE $\begin{smallmatrix} +12.0 \\ +7.0 \end{smallmatrix}$; PIN $\begin{smallmatrix} 0 \\ -3.5 \end{smallmatrix}$; CL $\begin{smallmatrix} +7.0 \\ +15.5 \end{smallmatrix}$
 LIMITS: HOLE $\frac{4.0120}{4.0070}$ PIN $\frac{4.0000}{3.9965}$ CL 0.0070 to 0.0155

7. PRECISION WITH WIDE TEMPERATURE VARIATIONS WOULD TYPICALLY CALL FOR RC3 OR RC4. RC2 PROBABLY TOO TIGHT FOR TEMP. CHANGE; RC5 PROBABLY TOO LOOSE FOR REQD PRECISION. RC3 OR RC4 NOT AVAILABLE IN TABLE B-6.

ILLUSTRATE LIMITS WITH RC5:

$$\text{HOLE: } \begin{matrix} +1.2 \\ -0 \end{matrix} ; \text{ PIN } \begin{matrix} -1.6 \\ -2.4 \end{matrix} ; \text{ CL } \begin{matrix} +1.6 \\ +3.6 \end{matrix}$$

$$\text{LIMITS: HOLE } \frac{0.7512}{0.7500} \quad \text{PIN } \frac{0.7487}{0.7476} \quad \text{CL } 0.0016 \text{ TO } 0.0036$$

8. LOOSE - RCB: HOLE $\begin{matrix} +4.0 \\ -0 \end{matrix}$; SHAFT $\begin{matrix} -5.0 \\ -7.5 \end{matrix}$; CL $\begin{matrix} +5.0 \\ +11.5 \end{matrix}$

ADJUST TOLERANCES FOR BASIC SHAFT SYSTEM - ADD 5.0.

$$\text{HOLE } \begin{matrix} +9.0 \\ +5.0 \end{matrix} ; \text{ SHAFT } \begin{matrix} 0 \\ -2.5 \end{matrix} ; \text{ CL } \begin{matrix} +5.0 \\ +11.5 \end{matrix}$$

$$\text{LIMITS: HOLE } \frac{1.5090}{1.5050} \quad \text{SHAFT } \frac{1.5000}{1.4975} \quad \text{CL } 0.0050 \text{ TO } 0.0115$$

10. $a = 0$; $b = 3.25/2 = 1.625 \text{ in}$; $c = 4.000/2 = 2.000 \text{ in}$: BOTH STEEL
USE FNS - HEAVY FORCE FIT $E = 30 \times 10^6 \text{ psi}$

$$\text{HOLE: } \begin{matrix} +2.2 \\ -0 \end{matrix} \quad \text{SHAFT: } \begin{matrix} +8.4 \\ +7.0 \end{matrix} \quad \text{INTERFERENCE: } \begin{matrix} 4.8 \\ 8.4 \end{matrix}$$

$$\text{LIMITS: HOLE } \frac{3.2522}{3.2500} \quad \text{SHAFT } \frac{3.2581}{3.2570} \quad \delta_{\text{MAX}} = 0.0084 \text{ in.}$$

$$(EQ. 13-2) \quad p = \frac{E\delta}{2b} \left[\frac{(c^2 - b^2)(b^2 - a^2)}{2b^2(c^2 - a^2)} \right] = \frac{(30 \times 10^6)(0.0084)}{2(1.625)} \left[\frac{(2.00^2 - 1.625^2)(1.625^2 - 0)}{2(1.625)^2(2.00^2 - 0)} \right]$$

$$(EQ. 13-4) \quad p = 13175 \text{ psi} \quad \sigma_o = p \left(\frac{c^2 + b^2}{c^2 - b^2} \right) = 13175 \left[\frac{2.00^2 + 1.625^2}{2.00^2 - 1.625^2} \right] = 64363 \text{ psi NOT ACCEPTABLE FOR 1020 HR; } \delta_{\text{MAX}} = 5.5 \times 10^{-3}$$

11. $a = 3.50/2 = 1.750 \text{ in}$; $b = 4.0/2 = 2.000 \text{ in}$; $c = 4.50/2 = 2.250 \text{ in}$

$$\text{FNS: HOLE } \begin{matrix} +1.4 \\ -0 \end{matrix} \quad \text{SHAFT } \begin{matrix} +4.9 \\ +4.0 \end{matrix} \quad \text{INT. } \begin{matrix} 2.6 \\ 4.9 \end{matrix}$$

$$\text{LIMITS: STEEL ID } = \frac{4.0014}{4.0000} ; \text{ BRONZE OD } = \frac{4.0049}{4.0040} ; \delta_{\text{MAX}} = 0.0049 \text{ in}$$

$$p = 1575 \text{ psi (EQ. 13-3)} \quad \text{USING } E_o = 30 \times 10^6 \text{ psi} \quad E_A = 15 \times 10^6 \text{ psi}$$

$$\sigma_o = 13438 \text{ psi (EQ. 13-4) INNER SURFACE OF STEEL SLEEVE} \quad v_o = 0.27 \quad v_A = 0.27$$

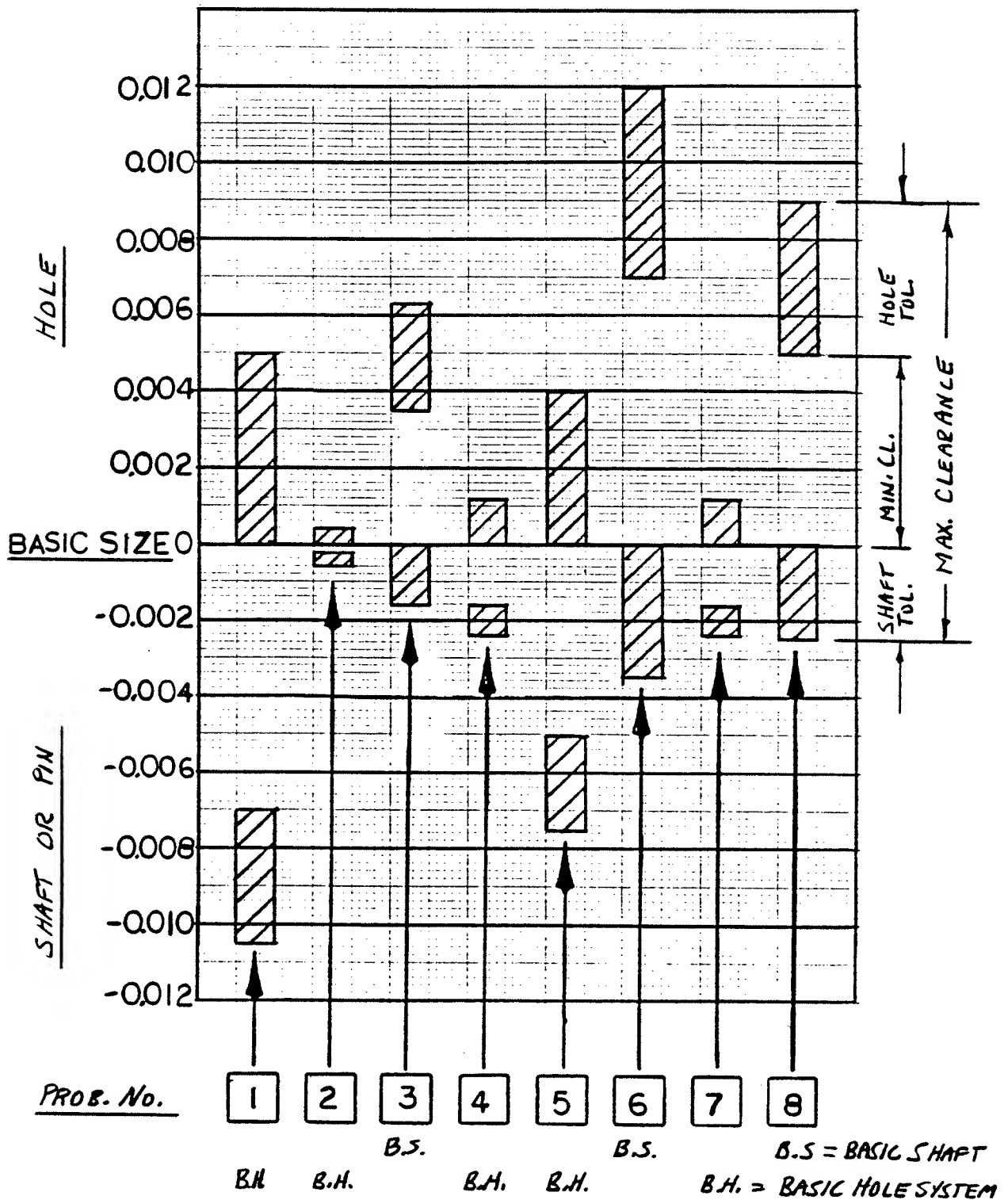
$$\sigma_A = -11869 \text{ psi (EQ. 13-5) OUTER SURFACE OF BRONZE BUSHING}$$

NOTE: APPENDIX A-12, BEARING BRONZE HAS A YIELD STRENGTH OF 18000 PSI.

$$N = \frac{S_y}{\sigma_A} = \frac{18000}{11869} = 1.52 \text{ LOW}$$

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TOLERANCE DIAGRAMS FOR PROBLEMS 1-8.



STRESSES FOR FORCE FITS Refer to Figure 13-6 for geometry	Problem identification: Problem 12	
Input Data:	Numerical values in <i>italics</i> must be inserted for each problem	
Inside radius of inner member =	$a =$	0.0000 in
Outside radius of inner member =	$b =$	1.5000 in
Outside radius of outer member =	$c =$	2.5000 in
Total interference =	$\delta =$	0.0072 in
Modulus of outer member =	$E_o =$	1.00E+07 psi ALUMINUM
Modulus of inner member =	$E_i =$	3.00E+07 psi STEEL
Poisson's ratio for outer member =	$\nu_o =$	0.33
Poisson's ratio for inner member =	$\nu_i =$	0.27
Computed results:		
Pressure at Mating Surface:	$p =$	8894 psi Using Equation (13-3)
Tensile Stress in the Outer Member:	$\sigma_o =$	18901 psi Using Equation (13-4)
Compressive Stress in the Inner Member:	$\sigma_i =$	-8894 psi Using Equation (13-5)
Increase in Diameter of Outer Member:	$\delta_o =$	0.0065 in Using Equation (13-6)
Decrease in Diameter of Inner Member:	$\delta_i =$	0.0065 in Using Equation (13-7)

EVALUATE FMS FIT FOR MAXIMUM INTERFERENCE.

HOLE TOLERANCE: +0.0018 SHAF: +0.0072 INTERFERENCE: 0.0092
-0 +0.0060 0.0072

MAX. INTERFERENCE = 0.0072 - 0 = 0.0072 SMALLEST HOLE
LARGEST SHAF

STRESS IN ALUMINUM = 18901 psi TENSION

ASSUMING NO ADDITIONAL LOADS AND STATIC CONDITION.

LET $N=2$. REQ'D $S_y = 2(18901 \text{ psi}) = 37802 \text{ psi}$

SPECIFY ANY ALUMINUM ALLOY WITH $S_y > 37800 \text{ psi}$

EXAMPLES:

2014-T4, $S_y = 42 \text{ ksi}$ OR 6061-T6, $S_y = 40 \text{ ksi}$

ANY STEEL FOR ROD WOULD BE SATISFACTORY, PROVIDED

$S_y > 2(8894 \text{ psi}) = 17788$

ANY CARBON OR ALLOY STEEL WITH $S_y > 18000 \text{ psi}$

FROM APP. 3. EXAMPLES AISI 1020 HR, $S_y = 30 \text{ ksi}$

AISI 1040 HR, $S_y = 42 \text{ ksi}$

13

STEEL SLEEVE ON ALUMINUM TUBE

$$a = [2.00 - 2(0.065)]/2 = 0.935 \text{ IN}; b = 2.00/2 = 1.000 \text{ IN}; c = 3.00/2 = 1.500 \text{ IN}$$

$$\text{EQ. 13-5 } \sigma_i = -p \left(\frac{b^2 + a^2}{b^2 - a^2} \right) = -p \left(\frac{1.00 + .8742}{1.00 - .8742} \right) = -p(14.898)$$

$$\text{FOR } \sigma_i = -8500 \text{ PSI}; p = \frac{8500 \text{ PSI}}{14.898} = 570 \text{ PSI MAX ALLOWABLE}$$

FROM EQ 13-3 SOLVE FOR δ

$$\delta_{\text{MAX}} = p \left\{ 2b \left[\frac{1}{E_o} \left(\frac{c^2 + b^2}{c^2 - b^2} + 1 \right) + \frac{1}{E_i} \left(\frac{b^2 + a^2}{b^2 - a^2} - \nu_i \right) \right] \right\}$$

$$E_o = 30 \times 10^6 \text{ PSI}; E_i = 10 \times 10^6 \text{ PSI}; \nu_o = 0.27; \nu_i = 0.33$$

$$\delta_{\text{MAX}} = 0.00177 \text{ IN.}$$

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NOMINAL DIA. = 3.250 IN.; ASSUME MAX INTERFERENCE = 0.0084 IN

FOR FINAL CLEARANCE = 0.002, CHANGE IN DIA. = 0.0084 + 0.002 = 0.0104

$$\Delta C = \frac{\delta}{\alpha L} = \frac{0.0104 \text{ IN}}{(6.5 \times 10^{-6})^\circ \text{F}^{-1} (3.250 \text{ IN})} = 492^\circ \text{F}; T_f = 75^\circ + 492 = 567^\circ \text{F}$$

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BRONZE - SHRINK FROM 75 TO -20; $\Delta T = 95^\circ \text{F}$

$$\delta = \alpha L (\Delta T) = (1.0 \times 10^{-6}) (4.00) (-95) = -0.0038 \text{ IN.}$$

MAX INTERFERENCE = 0.0049 IN.

$$\frac{-0.0038}{0.0011}$$

$$\frac{+0.0040 \text{ DESIRED CLEARANCE}}{0.0051 \text{ IN REQ'D EXPANSION OF STEEL.}}$$

$$\Delta T = \frac{\delta}{\alpha L} = \frac{0.0051}{(6.1 \times 10^{-6}) (4.000)} = 209^\circ \text{F}$$

$$\frac{75^\circ \text{F AMBIENT}}{284^\circ \text{F REQ'D STEEL TEMP.}}$$

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EQ. 13-7

$$\sigma_i = \frac{-2b p}{E_i} \left[\frac{b^2 + a^2}{b^2 - a^2} - \nu_i \right] = \frac{-2(2.00)(1575)}{(17 \times 10^6)} \left[\frac{2.00^2 + 1.75^2}{2.00^2 - 1.75^2} - 0.27 \right]$$

$$\sigma_i = -0.00269 \text{ IN}$$

$$\text{FINAL ID} = 3.500 - 0.00269 = 3.4973 \text{ IN.}$$

CHAPTER 14

ROLLING CONTACT BEARINGS

1. EQ. 14-2: $L_d = \left(\frac{C}{P_d}\right)^k (10^6) = \left(\frac{2350}{1675}\right)^{3.0} (10^6) = 2.76 \times 10^6 \text{ REV.}$

2. $L_d = (20,000 \text{ HR})(880 \text{ RPM})(60 \text{ MIN/HR}) = 1.06 \times 10^9 \text{ REV.}$

EQ. 14-3: $C = P_d (L_d / 10^6)^{1/k} = 1250 (1.06 \times 10^9 / 10^6)^{1/3} = 12,745 \text{ LB}$

3. EQ. 14-2 (a) $L_d = \left(\frac{C}{P_d}\right)^k (10^6) = \left(\frac{3150}{2200}\right)^3 (10^6) = 2.94 \times 10^6 \text{ REV.}$

(b) $L_d = \left(\frac{3150}{4500}\right)^3 (10^6) = 0.343 \times 10^6 = 3.43 \times 10^5 \text{ REV.}$

4. USE $L_d = (5000 \text{ HR})(1150 \text{ RPM})(60 \text{ MIN/HR}) = 1.04 \times 10^9 \text{ REV}$

$C = P_d (L_d / 10^6)^{1/k} = 1450 (1.04 \times 10^9 / 10^6)^{1/3} = 14,667 \text{ LB}$

5. FROM FIGURE 12-12:

REACTION AT B: $R_B = \sqrt{R_{Bx}^2 + R_{By}^2} = \sqrt{458^2 + 4620^2} = 4643 \text{ LB}$

REACTION AT D: $R_D = \sqrt{R_{Dx}^2 + R_{Dy}^2} = \sqrt{1223^2 + 1680^2} = 2078 \text{ LB}$

FROM EX. 12-1; DIA. AT B (MIN.) = $D_3 = 3.55 \text{ IN}$

DIA. AT D (MIN.) = $D_6 = 1.09 \text{ IN}$

INDUSTRIAL BLOWER; USE $L_d = (10,000 \text{ HR})(600 \text{ RPM})(60) = 3.60 \times 10^8 \text{ REV.}$

REQ'D. C VALUE AT B: $C = R_B (3.6 \times 10^8 / 10^6)^{1/3} = 4643 (7.114) = 33,029 \text{ LB.}$

AT D: $C = R_D (3.6 \times 10^8 / 10^6)^{1/3} = 2078 (7.114) = 14,782 \text{ LB.}$

FROM TABLE 14-3: BRG. 6319 HAS $C = 34,397 \text{ LB}$; BORE = 3.7432 IN.

BRG. 6311 HAS $C = 16,076 \text{ LB}$; BORE = 2.1654 IN.

6. DATA OF EX. PROB. 12-2 : FROM FIG. 12-16
 $R_B = \sqrt{589^2 + 164^2} = 611 \text{ LB}$; $R_D = \sqrt{393^2 + 188^2} = 436 \text{ LB}$
 $D_{\text{MIN}} = 2.02 \text{ IN}$; $D_{\text{OMN}} = 1.98 \text{ IN}$. FROM TABLE 12-2.

TABLE 14-4 : AGRICULTURAL EQ. - LBT $L_d = 5000 \text{ HR}$

$$L_d = (5000)(1700 \text{ RPM})(60) = 5.1 \times 10^8 \text{ REV}$$

$$\text{REQ'D } C \text{ VALUE AT B: } C = 611 \left(\frac{5.1 \times 10^8}{10^6} \right)^{1/3} = 4882 \text{ LB}$$

$$\text{AT D: } C = 436 \left(\frac{5.1 \times 10^8}{10^6} \right)^{1/3} = 3483 \text{ LB}$$

ATB: FROM TABLE 14-3 : BEARING 6011 HAS $C = 6317 \text{ LB}$ AND
 A BORE OF 2.1654 IN. C IS HIGHER THAN REQ'D BUT
 SHAFT DIA. MUST BE $> 2.02 \text{ IN}$.
 SPECIFY BRG. 6011 FOR BOTH B AND D.

7. DATA OF EX. PROB. 12-3 AND FIGS. 12-17 AND 12-18

$$R_A = \sqrt{507^2 + 41^2} = 509 \text{ LB} ; R_C = \sqrt{1697^2 + 393^2} = 1742 \text{ LB RADIAL}$$

R_A PURELY RADIAL ; BRG. C CARRIES 265 LB THRUST LOAD

$$D_{\text{MIN}} = 0.59 \text{ IN} ; D_{\text{CMN}} = 2.26 \text{ IN}$$

$$\text{USE } L_d = (20000 \text{ HR})(101 \text{ RPM})(60) = 1.2 \times 10^8 \text{ REV.}$$

$$\text{REQ'D. } C \text{ VALUE AT A: } C_A = 509 \left(\frac{1.2 \times 10^8}{10^6} \right)^{1/3} = 2519 \text{ LB}$$

BRG. 6302 HAS $C = 2563 \text{ LB}$ AND A BORE OF 0.5906 IN.

SHOULD BE COMPATIBLE WITH DIA. D_2 (FIG. 12-16) TO
 PROVIDE A SHOULDER FOR THE BEARING. A LIGHTER BEARING
 WITH A LARGER BORE MAY BE PREFERRED.

BEARING C : COMBINED RADIAL & THRUST LOAD. (EQ. 14-5)

$$\text{ASSUME } Y = 1.5 ; P = (1.0)(0.56)(1742) + (1.5)(265) = 1373 \text{ LB.}$$

$$C_c = 1373 \left(\frac{1.2 \times 10^8}{10^6} \right)^{1/3} = 6772 \text{ LB.}$$

BRG. 6212 HAS $C = 10679 \text{ LB}$, BORE = 2.3622 IN, $C_0 = 7307 \text{ LB}$

$$\text{CHECK: } T/C_0 = 265/7307 = 0.0363 \rightarrow C = 0.24$$

$$T/R = 265/1742 = 0.152 < e \text{ - USE EQ. 14-5; } P = 1.0 R_c = 1742 \text{ LB}$$

$$C_c = 1742 \left(\frac{1.2 \times 10^8}{10^6} \right)^{1/3} = 8592 \text{ LB} \rightarrow \text{BRG. 6212 OK,}$$

ROLLING CONTACT BEARING DESIGN CALCULATIONS - CHAPTER 14										Summary of Design Calculations									
USING DATA FROM TABLE 14-3										INNER RACE ROTATES IN ALL CASES					See manual solutions for details of calculations				
PROB	RADIAL LOAD, R (LB)	THRUST LOAD, T (LB)	SPEED (RPM)	LIFE (HR)	LIFE (REV.)	EQUIV LOAD, P (LB)	DYNAMIC LOAD, C (LB)	BRG.	RATED LOAD, C (LB)	BEARING BORE (mm)	BORE (in)	NS = Not specified							
NO. & BRG								NO.			MIN.	X	Y	C _o					
5-BRG. B	4643	0	600	10000	3.60E+08	4643	33029	6319	34397	95	3.7402	3.55	1	0	48561				
5-BRG. C	2078	0	600	10000	3.60E+08	2078	14782	6311	16076	55	2.1654	1.09	1	0	48561				
6-BRG. B	611	0	1700	5000	5.1E+08	611	4882	6011	6317	55	2.1654	2.00	1	0	4766				
6-BRG. D	436	0	1700	5000	5.1E+08	436	3483	6011	6317	55	2.1654	1.98	1	0	4766				
7-BRG. A	509	0	101	20000	1.21E+08	509	2519	6302	2563	15	0.5906	0.59	1	0	1214				
7-BRG. C	1742	265	101	20000	1.21E+08	1742	8621	6212	10679	60	2.3622	2.26	1	0	7307				
9	455	0	1150	20000	1.38E+09	455	5066	6306	6317	30	1.1811	NS	1	0	10				
10	857	0	450	30000	8.1E+08	857	7989	6308	9218	40	1.5748	NS	1	0	10				
11	1265	645	210	5000	6.30E+07	1579	6284	6307	7464	35	1.3780	NS	0.56	1.35	4272				
12	235	88	1750	20000	2.1E+09	301	3860	6305	5058	25	0.9843	NS	0.56	1.93	2608				
13	2875	1350	600	15000	5.4E+08	3919	31909	6318	32149	90	3.5433	NS	0.56	1.71	24281				
14(M-->lb)	854	0	3450	15000	3.11E+09	854	12459	6310	13894	50	1.9685	NS	1	0	10				
14(kN)	3.8	0	3450	15000	3.11E+09	3.80	55.44	6310	61.8	50	1.9685	NS	1	0	10				
15(kN)	5.6	2.8	450	2000	5.40E+07	6.78	25.61	6306	28.1	30	1.1811	NS	0.56	1.3	16				
16(kN)	10.5	0	1150	20000	1.38E+09	10.50	116.90	6316	124.0	80	3.1496	NS	1	0	10				
16(M-->lb)	2361	0	1150	20000	1.38E+09	2361	26286	6316	27878	80	3.1496	NS	1	0	10				
17(kN)	1.2	0.85	860	20000	1.03E+09	2.04	20.62	6305	22.5	25	0.9843	NS	0.56	1.61	11.6				
24-1	1750	350	101		0	1750	0	6211	9802	55	2.1654	NS	1	0	6520				
24-2	600	250	101		0	809	0	6211	9802	55	2.1654	NS	0.56	1.89	6520				
24-3	280	110	101		0	403	0	6211	9802	55	2.1654	NS	0.56	2.24	6520				
See manual solution for combined loading and overall life for Problem 24.																			

19.

VARYING LOADS : (BRG. 6324) $n = 600 \text{ rpm}$; $C = 46783$

$$\begin{array}{l} 1. \quad 4500 \text{ LB} \quad 25 \text{ MIN} \\ 2. \quad 2500 \text{ LB} \quad 15 \text{ MIN} \end{array} \quad F_m = \left(\frac{25(4500)^3 + 15(2500)^3}{40} \right)^{1/3} = 3975 \text{ LB}$$

$$L = \left(\frac{C}{F_m} \right)^3 = \left(\frac{46783}{3975} \right)^3 = 1630 \times 10^6 \text{ rev} \times \frac{\text{MIN}}{600 \text{ rev}} \times \frac{h}{60 \text{ MIN}} = 452.85 \text{ h}$$

20.

BEARING 6314, $n = 600 \text{ rpm}$; $C = 23381 \text{ LB}$.

$$\begin{array}{l} 1. \quad 2500 \text{ LB} \quad 25 \text{ MIN} \\ 2. \quad 1500 \text{ LB} \quad 15 \text{ MIN} \end{array} \quad F_m = \left(\frac{25(2500)^3 + 15(1500)^3}{40} \right)^{1/3} = 2226 \text{ LB}$$

$$L = \left(\frac{C}{F_m} \right)^3 = \left(\frac{23381}{2226} \right)^3 = 1159 \times 10^6 \text{ rev} \times \frac{h}{(600 \text{ rev/min})(60 \text{ MIN/h})} = 32.189 \text{ h}$$

21.

BEARING 6209, $n = 1700 \text{ rpm}$, $C = 7464 \text{ LB}$

$$\begin{array}{l} 1. \quad 600 \text{ LB} \quad 480 \text{ MIN} \\ 2. \quad 200 \text{ LB} \quad 115 \text{ MIN} \\ 3. \quad 100 \text{ LB} \quad 45 \text{ MIN} \end{array}$$

640 MIN

$$F_m = \left(\frac{480(600)^3 + 115(200)^3 + 45(100)^3}{640} \right)^{1/3} = 547 \text{ LB}$$

$$L = \left(\frac{C}{F_m} \right)^3 = \left(\frac{7464 \text{ LB}}{547 \text{ LB}} \right)^3 = \frac{2580 \times 10^6 \text{ rev}}{(1700 \text{ rev/min})(60 \text{ MIN/h})} = 24.909 \text{ h}$$

22.

BEARING 6209, $n = 1700 \text{ rpm}$, $C = 7464 \text{ LB}$

$$\begin{array}{l} 1. \quad 450 \text{ LB} \quad 480 \text{ MIN} \\ 2. \quad 180 \text{ LB} \quad 115 \text{ MIN} \\ 3. \quad 50 \text{ LB} \quad 45 \text{ MIN} \end{array}$$

640 MIN

$$F_m = \left(\frac{480(450)^3 + 115(180)^3 + 45(50)^3}{640} \right)^{1/3} = 411 \text{ LB}$$

$$L = \left(\frac{C}{F_m} \right)^3 = \left(\frac{7464}{411} \right)^3 = \frac{5989 \times 10^6 \text{ rev}}{(1700 \text{ rev/min})(60 \text{ MIN/h})} = 58.720 \text{ h}$$

23. BEARING 6205, $n = 101 \text{ rpm}$, $C = 3147 \text{ LB}$

1. 500 LB 6.75 h

2. 800 LB 0.40 h

3. 100 LB 0.85 h

8.00 h

$$F_m = \left(\frac{6.75(500)^3 + 0.40(800)^3 + 0.85(100)^3}{8.0} \right)^{1/3} = 508 \text{ LB}$$

$$L = \left(\frac{C}{F_m} \right)^3 = \left(\frac{3147}{508} \right)^3 = \frac{237.7 \times 10^6 \text{ rev}}{(101 \text{ rev/min})(60 \text{ min/h})} = 39231 \text{ h}$$

24. BEARING 6211, $n = 101 \text{ rpm}$, $C = 9802 \text{ LB}$, $C_0 = 6520 \text{ LB}$

COMBINED RADIAL AND THRUST LOADS:

COMPUTE EQUIVALENT LOAD P AS IN SECTION 14-10.

	$\frac{R}{1750 \text{ LB}}$	$\frac{T}{350 \text{ LB}}$	$\frac{T/R}{0.20}$	$\frac{T/C_0}{0.653}$	$\frac{e}{0.26}$	$\frac{Y}{1}$	$\frac{P}{1750 \text{ LB}} = R$
1. 6.75 h	1750 LB	350 LB	0.20	0.653	0.26	1	1750 LB
2. 0.40 h	600 LB	250 LB	0.417	0.0383	0.234	1.89	809 LB
3. 0.85 h	280 LB	110 LB	0.393	0.017	0.20	2.24	403 LB
8.00 h							

$$[P = 0.56 R + Y T] \rightarrow$$

USE EQUIVALENT LOADS TO COMPUTE F_m :

$$F_m = \left(\frac{6.75(1750)^3 + 0.40(809)^3 + 0.85(403)^3}{8.00} \right)^{1/3} = 1658 \text{ LB}$$

$$L = \left(\frac{C}{F_m} \right)^3 = \left(\frac{9802}{1658} \right)^3 = \frac{206.7 \times 10^6 \text{ rev}}{(101 \text{ rev/min})(60 \text{ min/h})} = 34,115 \text{ h}$$

25.

$$P_d = 1450 \text{ LB. } m = 1150 \text{ RPM. } L_d = 15000 \text{ h. } R = 0.95 \Rightarrow C_R = 0.62$$

$$\text{ACTUAL } L_d = (15000 \text{ h})(1150 \text{ RPM})(60 \text{ MIN/H}) = 1.04 \times 10^9 \text{ REV}$$

$$\text{ADJUSTED } L_{da} = L_d / C_R = 1.04 \times 10^9 / 0.62 = 1.68 \times 10^9 \text{ REV}$$

$$\text{EQ 14-3: } C = P_d \left(\frac{L_{da}}{10^6} \right)^{1/3} = 1450 \left(\frac{1.68 \times 10^9}{10^6} \right)^{1/3} = \underline{17229 \text{ LB.}}$$

26.

$$P_d = 509 \text{ LB. } m = 101 \text{ RPM. } L_d = 26000 \text{ h. } R = 0.99 \Rightarrow C_R = 0.21$$

$$\text{ACTUAL } L_d = (20000)(101)(60) = 1.21 \times 10^8 \text{ REV.}$$

$$\text{ADJUSTED } L_{da} = L_d / C_R = 1.21 \times 10^8 / 0.21 = 5.77 \times 10^8 \text{ REV}$$

$$\text{EQ. 14-3: } C = 509 \left(\frac{5.77 \times 10^8}{10^6} \right)^{1/3} = \underline{4238 \text{ LB}}$$

27.

$$P_d = 436 \text{ LB. } m = 1700 \text{ RPM. } L_d = 5000 \text{ h. } R = 0.97 \Rightarrow 0.44$$

$$\text{ACTUAL } L_d = (5000)(1700)(60) = 5.10 \times 10^8 \text{ REV.}$$

$$\text{ADJUSTED } L_{da} = L_d / C_R = 5.10 \times 10^8 / 0.44 = 1.16 \times 10^9 \text{ REV}$$

$$\text{EQ. 14-3: } C = 436 \left(\frac{1.16 \times 10^9}{10^6} \right)^{1/3} = \underline{4580 \text{ LB}}$$

28.

$$P_d = 1250 \text{ LB. } m = 880 \text{ RPM. } L_d = 20000 \text{ h. } R = 0.95 \Rightarrow C_R = 0.62$$

$$\text{ACTUAL } L_d = (20000 \text{ h})(880 \text{ RPM})(60 \text{ MIN/H}) = 1.06 \times 10^9 \text{ REV}$$

$$\text{ADJUSTED } L_d = L_d / C_R = 1.06 \times 10^9 / 0.62 = 1.70 \times 10^9 \text{ REV}$$

$$\text{EQ 14-3: } C = 1250 \left(\frac{1.70 \times 10^9}{10^6} \right)^{1/3} = \underline{14928 \text{ LB}}$$

CHAPTER 16

PLAIN SURFACE BEARINGS

All of the problems in this chapter are design problems with no unique solutions. A sample of each type of design problem is shown here.

1. $F = 225 \text{ LB}; D = 3.00 \text{ IN}; M = 1750 \text{ RPM}$ BOUNDARY LUBRICATED
 LET $L = 1.5D = 1.5(3.00) = 4.50 \text{ IN.}$ BEARINGS
 $p = \frac{F}{LD} = \frac{225 \text{ LB}}{(4.50)(3.00) \text{ IN}^2} = 16.67 \text{ psi}$
 $V = \pi D M / 12 = \pi(3.00)(1750) / 12 = 1374 \text{ FT/min}$
 $pV = (16.67)(1374) = 22900 \text{ psi} \cdot \text{ft/min}$
 REQ'D pV -RATING = $2(pV) = 2(22900) = 45800 \text{ psi} \cdot \text{ft/min}$
POROUS BRONZE BEARING MATERIAL / OIL IMPREGNATED
OR BU OR DU DRY LUBRICATED BEARING
4. $F = 75 \text{ LB}; D = 0.50 \text{ in}; M = 600 \text{ rpm}$
 LET $L = 1.5D = 1.5(0.50) = 0.75 \text{ in}$
 $p = F / LD = 75 / (0.75 \times 0.50) = 200.0 \text{ psi}$ } $pV = 15708 \text{ psi} \cdot \text{ft/min}$
 $V = \pi(0.50)(600) / 12 = 78.5 \text{ ft/min}$
 REQ'D $pV = 2(15708) = 31416 \text{ psi} \cdot \text{ft/min}$ POROUS BRONZE
OR BU BEARING
7. $F = 800 \text{ LB}; D = 3.00 \text{ in}; M = 350 \text{ rpm}$
 LET $L = 1.5D = 1.5(3.00) = 4.50 \text{ in}$
 $p = F / LD = 800 / (4.50)(3.00) = 59.3 \text{ psi}$ } $pV = 16290 \text{ psi} \cdot \text{ft/min}$
 $V = \pi(3.00)(350) / 12 = 275 \text{ ft/min}$
 REQ'D $pV = 2(16290) = 32580 \text{ psi} \cdot \text{ft/min}$ POROUS BRONZE
OR BU BEARING
8. $F = 60 \text{ LB}; D = 0.75 \text{ IN}; M = 750 \text{ RPM}; \text{TRY } L = 1.25D = 1.25(0.75) = 0.938 \text{ IN}$
 LET $L = 1.00 \text{ IN.}; L/D = 1.00 / 0.75 = 1.33 \text{ OK}$
 $p = F / LD = 60 / (1.00 \times 0.75) = 80 \text{ psi}$ } $pV = 11784 \text{ psi} \cdot \text{ft/min}$
 $V = \pi D M / 12 = \pi(0.75)(750) / 12 = 147.3 \text{ ft/min}$
 REQ'D pV RATING = $2(11784) = 23568 \text{ psi} \cdot \text{ft/min}$ USE BABBITT-HIGH TIN

HYDRODYNAMIC LUBRICATION

9. $F = 1250 \text{ LB}$; $D_{\text{MIN}} = 2.60 \text{ IN.}$; $n = 1750 \text{ RPM}$; ELECTRIC MOTOR

LET $D = 2.75 \text{ IN.}$, $R = D/2 = 1.375 \text{ IN.}$

FOR $\mu = 300 \text{ PSI}$; $L = \frac{F}{\mu D} = \frac{1250 \text{ LB}}{(300 \text{ LB/IN}^2)(2.75 \text{ IN})} = 1.515 \text{ IN}$

$L/D = 1.515/2.75 = 0.55$; LET $L = 0.5D = 1.375 \text{ IN.}$; $L/D = 0.50$

$\mu = \frac{F}{LD} = \frac{1250 \text{ LB}}{(1.375)(2.75) \text{ IN}^2} = 331 \text{ PSI OK}$

$C_d = 0.0036 \text{ IN}$; $C_n = 0.0018 \text{ IN}$; $R/C_n = 1.375/0.0018 = 754$
(FIG 16-3)

SURFACE FINISH: 16-32 μIN AVG

$h_o \approx 0.00025(2.75 \text{ IN}) = 0.00069 \text{ IN}$; USE $h_o = 0.0007 \text{ IN.}$

$h_o/C_n = 0.0007/0.0018 = 0.389 \rightarrow S = 0.29$
(FIG. 16-7)

$n_s = n/60 = 1750/60 = 29.17 \text{ REV/S}$

REQ'D $\mu = \frac{S \mu}{n_s (R/C_n)^2} = \frac{(0.29)(331)}{(29.17)(754)^2} = 5.79 \times 10^{-6} \text{ REYNS}$

SAE 50 OIL HAS $\mu = 6.5 \times 10^{-6} \text{ REYNS @ } 160^\circ\text{F}$

S IS PROPORTIONAL TO μ : $S = 0.29(6.5/5.79) = 0.326$

FROM FIG 16-8: $\xi(R/C_n) = 8.0$

$\xi = \frac{\xi(R/C_n)}{(R/C_n)} = \frac{8.0}{754} = 0.0106$

$T_f = \xi F R = (0.0106)(1250)(1.375) = 18.2 \text{ LB}\cdot\text{IN}$

$P_f = \frac{T_f n}{63000} = \frac{(18.2 \text{ LB}\cdot\text{IN})(1750 \text{ RPM})}{63000} = 0.506 \text{ HP}$

13.

$F = 500 \text{ LB}$; $D_{\min} = 1.15 \text{ IN}$; $M = 2500 \text{ RPM}$; PRECISION SPINDLE

LET $D = 1.200 \text{ IN}$; $R = \frac{1}{2} = 0.600 \text{ IN}$.

$$\text{FOR } \mu = 200 \text{ PSI} ; L = \frac{F}{\mu D} = \frac{500 \text{ LB}}{(200 \text{ LB/IN}^2)(1.200 \text{ IN})} = 2.08$$

$$L/D = 2.08/1.200 = 1.73 ; \text{LET } L/D = 1.50 ; L = 1.50 D = (1.5)(1.2) = 1.800 \text{ IN}$$

$$\mu = \frac{F}{L D} = \frac{500}{(1.800)(1.200)} = 232 \text{ PSI OK}$$

$$C_d = 0.0014 \text{ IN} ; C_n = 0.0007 \text{ IN} ; R/C_n = 0.600/0.0007 = 857$$

SURFACE FINISH: 8-16 μIN AVG.

$$h_o \approx 0.00025(1.20) = 0.00030 \text{ IN} ; h_o/C_n = 0.0003/0.0007 = 0.429$$

$$S = 0.11 ; M_s = 2500/60 = 41.67 \text{ NR/S}$$

$$\text{REQ'D } \mu = \frac{S \mu}{M_s (R/C_n)^2} = \frac{(0.11)(232)}{(41.67)(857)^2} = 0.832 \times 10^{-6} \text{ REYNS}$$

SAE 5W HAS $\mu = 0.91 \times 10^{-6}$ @ 160°F

$$S \text{ IS PROPORTIONAL TO } \mu : S = 0.11 \left(\frac{.91}{.832} \right) = 0.120$$

$$f(R/C_n) = 2.80 \text{ FROM FIG. 16-8:}$$

$$f = \frac{f(R/C_n)}{(R/C_n)} = \frac{2.80}{857} = 0.0033$$

$$T_f = f F R = (0.0033)(500 \text{ LB})(0.60 \text{ IN}) = 0.98 \text{ LB}\cdot\text{IN}$$

$$P_f = T_f M / 63000 = (0.98 \text{ LB}\cdot\text{IN})(2500 \text{ RPM}) / 63000 = 0.039 \text{ HP}$$

16.

$$F = 18.7 \text{ kN}; D_{\text{MIN}} = 100 \text{ mm}; n = 500 \text{ RPM}; \text{CONVEYOR}$$

$$D = 100 \text{ mm}; R = 50 \text{ mm}$$

$$\text{FOR } p = 2.0 \text{ MPa} = 2.0 \text{ N/mm}^2; L = \frac{F}{pD} = \frac{18.7 \times 10^3 \text{ N}}{(2.0 \text{ N/mm}^2) 100 \text{ mm}} = 93.5 \text{ mm}$$

$$\text{LET } L/D = 1.0; L = D = 100 \text{ mm}$$

$$p = \frac{F}{LD} = \frac{18.7 \times 10^3 \text{ N}}{(100)(100) \text{ mm}^2} = 1.87 \text{ N/mm}^2 = 1.87 \text{ MPa OK}$$

$$C_d = 0.15 \text{ mm}; C_L = 0.075 \text{ mm}; R/C_L = 50/0.075 = 667$$

(LARGE CLEARANCE DESIRED)

$$\text{SURFACE FINISH: NOTE: } 1.0 \mu\text{m} = 1.0 \times 10^{-6} \text{ m} \times \frac{0.0254 \text{ m}}{1 \text{ IN}} = 0.0254 \mu\text{m}$$

$$\text{THEN } 0.20 \mu\text{m}; 16 \mu\text{m} = 0.40 \mu\text{m}; 32 \mu\text{m} = 0.80 \mu\text{m}$$

$$63 \mu\text{m} = 1.60 \mu\text{m}$$

$$\text{SPECIFY SURFACE FINISH} = 0.8 \text{ TO } 1.6 \mu\text{m AVG.}$$

$$h_0 \approx 0.00025(100) = 0.025 \text{ mm}; h_0/C_L = 0.025/0.075 = 0.333$$

$$S = 0.096; M_s = n/60 = 500/60 = 8.33 \text{ REV/S}$$

$$\text{REQ'D } \mu = \frac{Sp}{M_s (R/C_L)^2} = \frac{(0.096)(1.87 \times 10^6 \text{ Pa})}{(8.33 \text{ REV/S})(667)^2} = 0.0485 \text{ Pa}\cdot\text{s}$$

$$\text{AT } 70^\circ\text{C, SAE 50 HAS } \mu = 0.046 \text{ Pa}\cdot\text{s}; h_0 \text{ SLIGHTLY } < 0.025 \text{ mm}$$

$$S = (0.096) \frac{0.046}{0.0485} = 0.091 \rightarrow f(R/C_L) = 2.6$$

$$f = \frac{f(R/C_L)}{R/C_L} = \frac{2.6}{667} = 0.0039$$

$$T_f = fFR = (0.0039)(18.7 \times 10^3 \text{ N})(50 \times 10^{-3} \text{ m}) = 3.65 \text{ N}\cdot\text{m}$$

$$P_f = T\dot{M} = 3.65 \text{ N}\cdot\text{m} \times 8.33 \frac{\text{REV}}{\text{S}} \times \frac{2\pi \text{ RAD}}{\text{REV}} = 191 \frac{\text{N}\cdot\text{m}}{\text{S}} = 191 \text{ WATTS}$$

17.

$$F = 2.25 \text{ kN}; D_{\text{min}} = 25 \text{ mm}; n = 2200 \text{ RPM}; \text{ MACHINE TOOL}$$

$$\text{LET } D = 25 \text{ mm}; R = D/2 = 12.5 \text{ mm} \quad \left| \begin{array}{l} C_d = 0.036 \text{ mm}; C_r = 0.018 \text{ mm} \\ R/C_r = \frac{12.5}{0.018} = 694 \end{array} \right.$$

$$\text{FOR } p = 2.0 \text{ MPa}; L = \frac{F}{p_0} = \frac{2.25 \times 10^3 \text{ N}}{(2.0 \text{ N/mm}^2)(25 \text{ mm})} = 45 \text{ mm}$$

$$\text{LET } L = 2D = 50 \text{ mm}; L/D = 2.0$$

$$p = \frac{F}{LD} = \frac{2.25 \times 10^3 \text{ N}}{(50)(25) \text{ mm}^2} = 1.80 \text{ N/mm}^2 = 1.80 \text{ MPa OK}$$

SURFACE FINISH: 16-32 μm AVG. (0.4 TO 0.8 μm AVG.)

$$l_0 \approx 0.00025(25) = 0.00625 \text{ mm} \approx 0.006 \text{ mm}$$

$$l_0/C_r = 0.006/0.018 = 0.333 \rightarrow S = 0.057$$

$$n_s = n/60 = 2200/60 = 36.7 \text{ REV/S}$$

$$\text{REQD. } \mu = \frac{S p}{n_s (R/C_r)^2} = \frac{(0.057)(1.80 \times 10^6 \text{ Pa})}{(36.7 \text{ REV/S})(694)^2} = 0.0058 \text{ Pa}\cdot\text{s}$$

SAE 5W HAS $\mu = 0.0066 \text{ Pa}\cdot\text{s}$ @ 70°C

$$S = 0.057 \left(\frac{0.0066}{0.0058} \right) = 0.065 \rightarrow f(R/C_r) = 1.6$$

$$f = \frac{f(R/C_r)}{R/C_r} = \frac{1.6}{694} = 0.0023$$

$$T_f = f F R = (0.0023)(2.25 \times 10^3 \text{ N})(12.5 \times 10^{-3} \text{ m}) = 0.065 \text{ N}\cdot\text{m}$$

$$P_f = T_f n = (0.065 \text{ N}\cdot\text{m})(36.7 \frac{\text{REV}}{\text{s}}) \left(\frac{2\pi \text{ RAD}}{\text{REV}} \right) = 15.0 \frac{\text{N}\cdot\text{m}}{\text{s}}$$

$$P_f = 15.0 \text{ WATTS}$$

19.

HYDROSTATIC LUBRICATION

$$F = 1250 \text{ LB}; p_s = 300 \text{ psi}; \text{LET } p_n = 250 \text{ psi}$$

$$\text{LET } R_n/R = 0.50; a_f = 0.55; q_f = 1.40$$

$$A_p = \frac{F}{a_f p_n} = \frac{1250 \text{ LB}}{(0.55)(250 \text{ LB/in}^2)} = 9.09 \text{ in}^2 = \pi D^2/4$$

$$D = \sqrt{4 A_p / \pi} = \sqrt{4(9.09) / \pi} = 3.40 \text{ in. OK}$$

$$R = D/2 = 1.70 \text{ in}; R_n = 0.5 R = 0.5(1.7) = 0.85 \text{ in.}$$

$$\text{LET } h = 0.005 \text{ in}; \text{SAE 30 OL @ } 120^\circ \text{F}; \mu = 8.3 \times 10^{-6} \text{ LB}\cdot\text{s/in}^2$$

$$Q = q_f \frac{F R^3}{A_p \mu} = \frac{(1.40)(1250 \text{ LB})(0.005)^3 \text{ in}^3}{(9.09 \text{ in}^2)(8.3 \times 10^{-6} \text{ LB}\cdot\text{s/in}^2)} = 2.90 \text{ in}^3/\text{s}$$

$$P = p_n Q = (250 \text{ LB/in}^2)(2.90 \text{ in}^3/\text{s}) = 725 \text{ LB}\cdot\text{in/s} \times \frac{1 \text{ hp}}{6600 \text{ LB}\cdot\text{in/s}} = 0.11 \text{ hp}$$

21.

$$F = 3500 \text{ LB}; p_s = 500 \text{ psi}; \text{LET } p_n = 350 \text{ psi}$$

$$\text{LET } R_n/R = 0.60; a_f = 0.62; q_f = 1.60$$

$$A_p = \frac{F}{a_f p_n} = \frac{3500 \text{ LB}}{(0.62)(350 \text{ LB/in}^2)} = 16.13 \text{ in}^2$$

$$D = \sqrt{4 A_p / \pi} = \sqrt{4(16.13) / \pi} = 4.53 \text{ in} - \text{USE } D = 4.50 \text{ in.}$$

$$A_p = \pi D^2/4 = \pi(4.50)^2/4 = 15.90 \text{ in}^2$$

$$p_n = \frac{F}{a_f A_p} = \frac{3500 \text{ LB}}{(0.62)(15.90 \text{ in}^2)} = 358 \text{ psi OK}$$

$$R = D/2 = 2.25 \text{ in.}; R_n = 0.60 R = 1.35 \text{ in.}$$

$$\text{LET } h = 0.008 \text{ in.}; \text{SAE 40 OL @ } 140^\circ \text{F}; \mu = 7.0 \times 10^{-6} \text{ LB}\cdot\text{s/in}^2$$

$$Q = \frac{q_f F R^3}{A_p \mu} = \frac{(1.60)(3500 \text{ LB})(0.008)^3 \text{ in}^3}{(15.90 \text{ in}^2)(7.0 \times 10^{-6} \text{ LB}\cdot\text{s/in}^2)} = 25.8 \text{ in}^3/\text{s}$$

$$P = p_n Q = (358 \text{ LB/in}^2)(25.8 \text{ in}^3/\text{s}) = \frac{9145 \text{ LB}\cdot\text{in/s}}{(6600 \text{ LB}\cdot\text{in/s})/\text{hp}} = 1.39 \text{ hp}$$

25. HYDROSTATIC LUBRICATION - METRIC DATA

$$F = 22.5 \text{ kN}; p_s = 2.0 \text{ MPa}; \text{LET } p_n = 1.60 \text{ MPa} = 1.6 \times 10^6 \frac{\text{N}}{\text{m}^2} = 1.60 \text{ N/mm}^2$$

$$R_n/R = 0.60; a_f = 0.62; q_f = 1.60$$

$$A_p = \frac{F}{a_f p_n} = \frac{22.5 \times 10^3 \text{ N}}{(62)(1.60 \text{ N/mm}^2)} = 2.27 \times 10^4 \text{ mm}^2$$

$$D = \sqrt{4A_p/\pi} = \sqrt{4(2.27 \times 10^4)/\pi} = 170 \text{ mm OK}$$

$$R = D/2 = 85 \text{ mm}; R_n = 0.6R = 51.0 \text{ mm}$$

$$\text{LET } h = 0.15 \text{ mm}; \text{ AT } 50^\circ\text{C, SAE 30 OIL HAS } \mu = 0.054 \text{ Pa}\cdot\text{s}$$

$$Q = \frac{q_f F h^3}{A_p \mu} = \frac{(1.60)(22500 \text{ N})(.15)^3 \text{ mm}^3}{(2.27 \times 10^4 \text{ mm}^2)(0.054 \text{ N}\cdot\text{s/m}^2)} \times \frac{1 \text{ m}}{10^3 \text{ mm}} = 9.91 \times 10^{-5} \text{ m}^3/\text{s}$$

$$P = p_n Q = \left(1.60 \times 10^6 \frac{\text{N}}{\text{m}^2}\right) \left(9.91 \times 10^{-5} \frac{\text{m}^3}{\text{s}}\right) = 159 \frac{\text{N}\cdot\text{m}}{\text{s}} = 159 \text{ WATTS}$$

26.

$$F = 1.20 \text{ kN}; p_s = 750 \text{ kPa} = 0.75 \text{ MPa}$$

$$\text{LET } p_n = 600 \text{ kPa} = 0.60 \text{ MPa} = 0.60 \times 10^6 \text{ N/m}^2 = 0.60 \text{ N/mm}^2$$

$$R_n/R = 0.60; a_f = 0.62; q_f = 1.60$$

$$A_p = \frac{F}{a_f p_n} = \frac{1.20 \times 10^3 \text{ N}}{(62)(.60 \text{ N/mm}^2)} = 3226 \text{ mm}^2$$

$$D = \sqrt{4A_p/\pi} = \sqrt{4(3226)/\pi} = 64 \text{ mm OK}$$

$$R = D/2 = 32 \text{ mm}; R_n = 0.6R = .6(32) = 19.2 \text{ mm}$$

$$\text{LET } h = 0.10 \text{ mm}; \text{ AT } 60^\circ\text{C, SAE 10W OL HAS } \mu = 0.014 \text{ Pa}\cdot\text{s}$$

$$Q = \frac{q_f F h^3}{A_p \mu} = \frac{(1.60)(1200 \text{ N})(.10)^3 \text{ mm}^3}{(3226 \text{ mm}^2)(0.014 \text{ N}\cdot\text{s/m}^2)} \cdot \frac{1 \text{ m}}{10^3 \text{ mm}} = 4.25 \times 10^{-5} \text{ m}^3/\text{s}$$

$$P = p_n Q = 0.60 \times 10^6 \frac{\text{N}}{\text{m}^2} \cdot 4.25 \times 10^{-5} \frac{\text{m}^3}{\text{s}} = 25.5 \frac{\text{N}\cdot\text{m}}{\text{s}} = 25.5 \text{ WATTS}$$

CHAPTER 17 LINEAR MOTION ELEMENTS

5.

$$\text{REQ'D. } A_t = \frac{F}{\sigma_u} = \frac{30000 \text{ LB}}{10000 \text{ LB/IN}^2} = 3.0 \text{ IN}^2 \quad \text{USE } 2\frac{1}{2}\text{-3 ACME THREAD.}$$

$$A_t = 3.802 \text{ IN}^2$$

6.

$$\text{REQ'D } A_s = \frac{F}{T_u} = \frac{30000 \text{ LB}}{6000 \text{ LB/IN}} = 5.0 \text{ IN}^2$$

FOR AN ACME 2 1/2-3: $A_s = 4.075 \text{ IN}^2/\text{IN OF LENGTH}$
 REQ'D. $L = 5.0 / 4.075 = 1.23 \text{ IN.}$

7.

$$\lambda = \tan^{-1} \left(\frac{L}{\pi D_p} \right) = \tan^{-1} \left(\frac{0.333}{\pi(2.2932)} \right) = \tan^{-1}(0.0463) = 2.65^\circ$$

$\uparrow \tan \lambda$

$$\cos \phi = \cos 14.5^\circ = 0.968$$

(EQ. 18-10)

$$T_u = \frac{F D_p}{2} \left[\frac{\cos \phi \tan \lambda + f}{\cos \phi - f \tan \lambda} \right] = \frac{30000(2.2932)}{2} \left[\frac{(0.968)(0.0463) + 0.15}{0.968 - 0.15(0.0463)} \right]$$

$$T_u = 6974 \text{ lb} \cdot \text{in}$$

8.

$$T_d = \frac{F D_p}{2} \left[\frac{f - \cos \phi \tan \lambda}{\cos \phi + f \tan \lambda} \right] = \frac{30000(2.2932)}{2} \left[\frac{0.15 - (0.968)(0.0463)}{0.968 + 0.15(0.0463)} \right]$$

$$T_d = 3712 \text{ lb} \cdot \text{in}$$

9.

SQUARE THREAD: 3/4-6; $F = 4000 \text{ LB}$, $L = p = 1/6 = 1/6 = 0.1667 \text{ IN.}$

$$T_u = \frac{F D_p}{2} \left[\frac{L + \pi f D_p}{\pi D_p - f L} \right] = \frac{4000(0.6424)}{2} \left[\frac{0.1667 + \pi(0.15)(0.6424)}{\pi(0.6424) - 0.15(0.1667)} \right]$$

$$T_u = 303 \text{ lb} \cdot \text{in}$$

10.

$$T_d = \frac{F D_p}{2} \left[\frac{\pi f D_p - L}{\pi D_p + f L} \right] = \frac{4000(0.6424)}{2} \left[\frac{\pi(0.15)(0.6424) - 0.1667}{\pi(0.6424) + 0.15(0.1667)} \right] = 86 \text{ lb} \cdot \text{in}$$

11.

$$\lambda = \tan^{-1} (L / \pi D_p) = \tan^{-1} (0.1667 / \pi(0.6424)) = 4.72^\circ$$

$\tan \lambda = 0.083 < f \rightarrow \text{SELF LOCKING}$

$$12. e = \frac{FL}{2\pi T_m} = \frac{(4000)(0.1667)}{2\pi(303)} = \underline{0.35 \text{ or } 35\%}$$

$$13. n = \frac{0.50 \text{ IN}}{S} \times \frac{1 \text{ REV.}}{0.1667 \text{ IN}} \times \frac{60 S}{\text{MIN}} = \underline{180 \text{ RPM}}$$

$$P = T_m / 63000 = (303)(180) / 63000 = \underline{0.866 \text{ hp}}$$

$$14. \text{TRAVEL} = \frac{2 \text{ IN}}{\text{CYCLE}} \times \frac{10 \text{ CYCLES}}{\text{HR}} \times \frac{24 \text{ HR}}{\text{DAY}} \times \frac{365 \text{ DAYS}}{\text{YR}} \times \frac{10 \text{ YRS}}{1} = 2.10 \times 10^7 \text{ INCHES}$$

AT 600 LB; 3/4-2 SCREW REQ'D.; $L = 0.50 \text{ IN}$

$$15. T = \frac{FL}{2\pi e} = \frac{(600)(.50)}{2\pi(1.9)} = \underline{53.1 \text{ lb-in}}$$

$$16. n = \frac{1 \text{ REV}}{0.50 \text{ IN}} \times \frac{10.0 \text{ IN}}{\text{MIN.}} = \underline{20.0 \text{ RPM}}$$

$$P = \frac{T_m}{63000} = \frac{53.1(20.0)}{63000} = \underline{0.017 \text{ hp.}}$$

17. FOR THE 3/4-2 SCREW AT 600 LB, TRAVEL LIFE = $1.04 \times 10^8 \text{ IN.}$

$$1.04 \times 10^8 \text{ IN.} \times \frac{\text{CYCLE}}{24 \text{ IN}} \times \frac{\text{HR}}{20 \text{ CYCLES}} \times \frac{\text{DAY}}{24 \text{ HR}} \times \frac{\text{YR}}{365 \text{ DAYS}} = \underline{24.7 \text{ YRS}}$$

METRIC TRAPEZOIDAL POWER SCREWS - TABLE 17-1 M

18. SPECIFY A SIZE: LOAD = 125 kN; $\sigma_a = 75 \text{ MPa} = \frac{F}{A_t}$	PROBLEMS 18 TO 23 USE SAME DATA
REQ'D. $A_t = \frac{F}{\sigma_a} = \frac{125000 \text{ N}}{75 \text{ N/mm}^2} = 1667 \text{ mm}^2$ - USE M55 X 9 SCREW $A_t = 1791 \text{ mm}^2$	

NOMINAL $D_o = 55 \text{ mm}$; LEAD = PITCH = 9.0 mm FOR SINGLE THREAD
 $D_p = 50.5 \text{ mm}$

19. FIND TORQUE TO RAISE 125 kN LOAD FOR $f = 0.15$.

$$T_u = \frac{F D_p}{2} \left[\frac{L + \pi f D_p}{\pi D_p - f L} \right] = \frac{125000 \text{ N}(50.5 \text{ mm})}{2} \left[\frac{9.0 + \pi(0.15)(50.5)}{\pi(50.5) - (0.15)(9.0)} \right] \quad \text{EQ 17-2}$$

$$T_u = 658559 \text{ N}\cdot\text{mm} = \underline{658.6 \text{ N}\cdot\text{m}}$$

20. FIND TORQUE TO LOWER LOAD. $F = 125 \text{ kN}$, $f = 0.15$

$$T_d = \frac{F D_p}{2} \left[\frac{\pi f D_p - L}{\pi D_p + f L} \right] = \frac{125000 \text{ N}(50.5 \text{ mm})}{2} \left[\frac{\pi(0.15)(50.5) - 9}{\pi(50.5) + (0.15)(9)} \right] \quad \text{EQ 17-1}$$

$$T_d = 291905 \text{ N}\cdot\text{mm} = \underline{291.9 \text{ N}\cdot\text{m}}$$

21. FIND POWER TO RAISE 125 kN - 4250 mm IN 7.5 s. M55 X 9 SCREW

$$P = T_u \cdot \omega. \quad T_u = 658.6 \text{ N}\cdot\text{m FROM PROB. 19.}$$

$$\omega = \frac{N}{L} = \frac{d/t}{L} = \frac{4250 \text{ mm}}{7.5 \text{ s}} \times \frac{1 \text{ REV}}{9 \text{ mm}} \times \frac{2\pi \text{ RAD}}{1 \text{ REV}} = 395.6 \text{ RAD/s}$$

$$P = T_u \cdot \omega = 658.6 \text{ N}\cdot\text{m} \times 395.6 \text{ RAD/s} = 260\,547 \text{ N}\cdot\text{m/s} = 260\,547 \text{ W}$$

(VERY HIGH)

$$P = 260.5 \text{ kW}$$

22. FIND LEAD ANGLE λ : $\lambda = \tan^{-1}\left(\frac{L}{\pi D_p}\right) = \tan^{-1}\left(\frac{9}{\pi(50.5)}\right) = 3.25^\circ$ EQ. 17-3
FOR $\lambda < 5^\circ$ - SELF LOCKING.

23. FIND EFFICIENCY: $e = \frac{FL}{2\pi T_u} = \frac{125000 \text{ N}(9 \text{ mm})}{2\pi(658.559 \text{ N}\cdot\text{m})} = 0.272; 27.2\%$

24. SPECIFY A POWER SCREW SIZE: $F = 8500 \text{ N}$; $\sigma_a = 110 \text{ MPa}$ PROBLEM 24-29
USE SAME DATA.

$$\text{REQD } A_T = \frac{F}{\sigma_a} = \frac{8500 \text{ N}}{110 \text{ N/mm}^2} = 77.3 \text{ mm}^2 \text{ --- M14 X 3 SCREW}$$

$$D_p = 12.5 \text{ mm}, \quad L = \phi = 3.0 \text{ mm} \quad A_T = 103.9 \text{ mm}^2$$

25. FIND TORQUE TO RAISE LOAD; $f = 0.15$

$$T_u = \frac{F D_p}{2} \left[\frac{L + \pi f D_p}{\pi D_p - f L} \right] = \frac{8500 \text{ N}(12.5 \text{ mm})}{2} \left[\frac{3.0 + \pi(0.15)(12.5)}{\pi(12.5) - 0.15(3)} \right]$$

$$T_u = 12\,167 \text{ N}\cdot\text{mm} = 12.167 \text{ N}\cdot\text{m}$$

26. FIND TORQUE TO LOWER LOAD. $F = 8500 \text{ N}$; $f = 0.15$

$$T_d = \frac{F D_p}{2} \left[\frac{\pi f D_p - L}{\pi D_p + f L} \right] = \frac{8500 \text{ N}(12.5 \text{ mm})}{2} \left[\frac{\pi(0.15)(12.5) - 3}{\pi(12.5) + 0.15(3)} \right]$$

$$T_d = 3956 \text{ N}\cdot\text{mm} = 3.956 \text{ N}\cdot\text{m}$$

27. FIND POWER TO RAISE LOAD: 240 mm IN 3.5 s.

$$P = T_u \cdot \omega; \quad T_u = 12.167 \text{ N}\cdot\text{m} \text{ [PROB 25]}$$

$$\omega = \frac{N}{L} = \frac{d/t}{L} = \frac{240 \text{ mm}}{3.5 \text{ s}} \times \frac{1 \text{ REV}}{3.0 \text{ mm}} \times \frac{2\pi \text{ RAD}}{1 \text{ REV}} = 143.62 \text{ RAD/s}$$

$$P = T_u \cdot \omega = 12.167 \text{ N}\cdot\text{m} (143.62 \text{ RAD/s}) = 1747 \text{ N}\cdot\text{m/s} = 1747 \text{ W} = 1.747 \text{ kW}$$

28. LEAD ANGLE $\lambda = \tan^{-1}\left(\frac{L}{\pi D_p}\right) = \tan^{-1}\left(\frac{3.0}{\pi(12.5)}\right) = 4.37^\circ$

29. EFFICIENCY = $e = \frac{FL}{2\pi T_u} = \frac{8500 \text{ N}(3.0 \text{ mm})}{2\pi(12167 \text{ N}\cdot\text{mm})} = 0.334 = 33.4\%$

30.

BALL SCREW FROM PROBLEM 14, $\frac{3}{4}-2$, LENGTH = 28.0 IN
FIND ESTIMATE OF CRITICAL SPEED. EQ. 17-15

$$M_c (\text{RPM}) = \frac{476 \times 10^6 d K_s}{(SF) L^2}$$

d = MINOR DIA. OF SCREW. THIS VALUE NOT AVAILABLE IN THIS BOOK, OR ON INTERNET SITE 10. AS AN ESTIMATE, USE MINOR DIA. FOR A $\frac{3}{4}$ ACME SCREW - $d \approx 0.55$ IN - FROM MACHINERY'S HANDBOOK, 28TH ED., P. 1830.

L = 28.0 IN - BETWEEN SINGLE BEARINGS AT ENDS

$$K_s = 1.00 \quad \checkmark$$

LET $SF = 1.0$ TO ESTIMATE CRITICAL SPEED.

$$M_c = \frac{476 \times 10^6 (0.55 \text{ IN}) (1.0)}{(1) (28.0 \text{ IN})^2} = 3339 \text{ RPM}$$

SAFE OPERATING SPEED - $SF = 3.0$

$$M_{\text{OPER}} = \frac{M_c}{SF} = \frac{3339 \text{ RPM}}{3} = 1113 \text{ RPM}$$

CHAPTER 18 SPRINGS

$$1 \quad k = \Delta F / \Delta L = 12.0 / (2.75 - 1.85) = \underline{13.3 \text{ LB/IN.}}$$

$$2 \quad k = \frac{F_1 - 0}{L_f - L_0} ; L_f - L_0 = \frac{F_0}{k} ; L_f = L_0 + \frac{F_0}{k} = 1.25 + \frac{4.65 \text{ LB}}{18.840/\text{IN.}} = \underline{1.497 \text{ IN.}}$$

$$3 \quad k = \frac{F_2 - F_0}{L_2 - L_0} ; F_2 = k(L_2 - L_0) + F_0 = 76.7(0.830 - 0.626) + 32.2 = \underline{47.8 \text{ LB} = F_2}$$

$$L_f = L_0 + F_1/k = 0.830 + 32.2/76.7 = \underline{1.25 \text{ IN} = L_f}$$

$$4 \quad k = \Delta F / \Delta L = 99.2 \text{ N} / (63.5 - 37.1) \text{ mm} = \underline{3.76 \text{ N/mm}}$$

$$5 \quad L_f = L_0 + F_1/k = 39.47 \text{ mm} + 54.05 \text{ N} / 1.47 \text{ N/mm} = \underline{76.24 \text{ mm}}$$

$$6 \quad F_2 = k(L_2 - L_0) + F_0 = 8.95(29.4 - 21.4) + 134 \text{ N} = \underline{205.6 \text{ N} = F_2}$$

$$L_f = L_0 + F_1/k = 29.4 + 134/8.95 = \underline{44.4 \text{ mm} = L_f}$$

$$7 \quad I D = O D - 2 D_w = 1.100 - 2(0.085) = \underline{0.93 \text{ IN} = I D}$$

$$D_m = O D - D_w = 1.100 - 0.085 = \underline{1.015 \text{ IN} = D_m}$$

$$C = D_m / D_w = 1.015 / 0.085 = \underline{11.94 = C}$$

$$L_s = N D_w ; N = L_s / D_w = 0.563 / 0.085 = \underline{6.6 \text{ COILS}}$$

$$8 \quad D_m = O D - D_w = 0.560 - 0.059 = 0.501 \text{ IN} ; C = D_m / D_w = 0.501 / 0.059 = \underline{8.49 = C}$$

$$L_f = p N_a + 2 D_w ; N_a = N - 2 = 19 - 2 = 17 \text{ ACTIVE COILS}$$

$$p = \frac{L_f - 2 D_w}{N_a} = \frac{4.22 - 2(0.059)}{17} = \underline{0.241 \text{ IN} = p}$$

$$\lambda = \tan^{-1} \left(\frac{p}{\pi D_m} \right) = \tan^{-1} \left(\frac{0.241}{\pi(0.501)} \right) = \underline{8.7 \text{ DEGREES} = \lambda}$$

$$L_s = N D_w = 19(0.059) = \underline{1.12 \text{ IN.} = L_s}$$

9 FROM EQ 19-6: $F_0 = \frac{F_s G D_w}{8 C^3 N_a} = \frac{(4.22 - 3.00)(11.85 \times 10^6)(0.059)}{8(8.49)^3(17)} = 10.25 \text{ LB} = F_s$

EQ. 7-4: $T_s = \frac{8 k F_s C}{\pi D_w^2} = \frac{8(117)(10.25)(8.49)}{\pi(0.059)^2} = 74500 \text{ PSI} = T_s$

$K = \frac{4C-1}{4C-4} + \frac{0.615}{C} = \frac{4(8.49)-1}{4(8.49)-4} + \frac{0.615}{8.49} = 1.17$

FROM FIGURE 18-9: MUSIC WIRE: $T_u = 132000 \text{ PSI}$ FOR AVG. SERVICE - OK.

10. $L_s/D_w = 4.22/0.501 = 8.42$: FROM FIG 19-15, CURVE A, $(F_0/L_s)_{CR} = 0.18$
 $(F_0)_{CR} = 0.18 L_s = 0.18(4.22) = 0.76 \text{ IN}$

ACTUAL $F_0 = 4.22 - 3.00 = 1.22 \text{ IN} > 0.76 \text{ IN}$ - BUCKLING SHOULD OCCUR

11. (EQ. 19-3)
 $(OD)_s = \sqrt{D_w^2 + \frac{F_s - D_w^2}{\pi^2}} + D_w = \sqrt{(0.501)^2 + \frac{0.241 - (0.059)^2}{\pi^2}} + 0.059 = 0.583 \text{ IN}$

12. $F_s = k(L_s - L_0) = 8.40(4.22 - 1.112) = 26.05 \text{ LB}$

FROM PROB. 9: $k = \frac{F_0}{L_s - L_0} = \frac{10.25}{4.22 - 3.00} = 8.40 \text{ LB/IN}$

$T_s = T_u \times F_s/F_0 = 74500 \text{ PSI} \times 26.05/10.25 = 189300 \text{ PSI}$ TOO HIGH
 FROM FIG 19-9, LIGHT SERVICE, $T_{su} = 147000 \text{ PSI} = \text{APPROX. } S_y$

Notes concerning Problems 13 - 35: Most of these problems are design problems. No single unique solutions exist. Sample solutions are shown.

Problems 13 through 24 are compression springs. Problem 13 is done by both methods 1 and 2 as outlined in the text. Others are done by one or the other method. In some problems only summary results are shown.

Problems 25 through 31 are extension springs. Problem 25 is worked out in detail. Problems 26 through 30 were designed with the aid of a computer program using the same procedure. Only summary results are shown. Problem 31 is the stress analysis of the ends of the spring.

Problems 32 through 35 are torsion springs. In each case, the design of each end is required. It was assumed that ends would be straight with lengths L_1 and L_2 as shown. The effects of the ends on the spring rate were then included in the analysis.

13 METHOD 1 $F_0 = 220 \text{ LB}$; $F_i = 180 \text{ LB}$; $\Delta F = 220 - 180 = 40 \text{ LB}$; $\Delta L = 0.50 \text{ IN}$.

STEP 1 - ASTM A229; SEVERE SERVICE; $G = 11.2 \times 10^6 \text{ PSI}$

STEPS 2-5 - LET $L_i = 3.00 \text{ IN}$; $D_m = 3.00 \text{ IN}$. (DESIGN DECISIONS)

$$R = \Delta F / \Delta L = 40 \text{ LB} / 0.50 \text{ IN} = 80 \text{ LB/IN}$$

$$L_f = L_i + F_i / R = 3.00 + 180 / 80 = 5.25 \text{ IN}$$

$$L_o = L_i - \Delta L = 3.00 - 0.50 = 2.50 \text{ IN}$$

$$F_o = L_f - L_o = 5.25 - 2.50 = 2.75 \text{ IN}$$

STEP 6 - ASSUME $T_d = 80,000 \text{ PSI}$

STEP 7 - (EQ. 18-7)

$$D_w = \left[\frac{(3.06)(F_o)(D_m)}{T_d} \right]^{1/3} = \left[\frac{3.06(220)(3.00)}{80,000} \right]^{1/3} = 0.293 \text{ IN}$$

STEP 8 - SELECT $D_w = 0.3065 \text{ IN}$ U.S. GAGE 0; $T_d = 78 \text{ KSI}$; $T_{max} = 104 \text{ KSI}$

STEP 9 - $C = D_m / D_w = 3.00 / 0.3065 = 9.79$; $K = 1.15$ (FIG. 18-14)

STEP 10 -

$$T_o = \frac{8 K F_o D_m}{\pi D_w^3} = \frac{8(1.15)(220)(3.00)}{\pi(0.3065)^3} = 67,126 \text{ PSI} \quad \text{OK}$$

$$\text{STEP 11 - } N_a = \frac{G D_w}{8 K C^3} = \frac{(11.2 \times 10^6)(0.3065)}{8(80)(9.79)^3} = 5.72 \text{ COILS}$$

$$\text{STEP 12 - } L_s = D_w(N_a + 2) = 0.3065(5.72 + 2) = 2.365 \text{ IN}$$

$$F_s = R(L_f - L_s) = 80(5.25 - 2.365) = 231 \text{ LB}$$

$$T_s = T_o(F_s / F_o) = 67,126(231 / 220) = 70,420 \text{ PSI} \quad \text{OK}$$

$$\text{STEP 13 - } D_o = D_m + D_w = 3.00 + 0.3065 = 3.3065 \text{ IN}$$

$$D_i = D_m - D_w = 3.00 - 0.3065 = 2.694 \text{ IN}$$

SUMMARY: $D_w = 0.3065 \text{ IN}$; $L_f = 5.25 \text{ IN}$; $L_o = 2.50 \text{ IN}$; $D_m = 3.00 \text{ IN}$.

13 METHOD 2 PROCESS STARTS SAME AS METHOD 1 WITHOUT D_m .

$$\text{STEP 2 - (EQ. 18-10)} \quad D_w = \sqrt[3]{21.4 F_o / T_d} = \sqrt[3]{21.4(220) / 80,000} = 0.243 \text{ IN}$$

STEP 3 - TRY $D_w = 0.2625 \text{ IN}$ U.S. GAGE 2; $T_d = 80,000 \text{ PSI}$

STEP 4 - (EQ. 18-11)

$$N_{a \max} = (L_o - 2 D_w) / D_w = [2.50 - 2(0.2625)] / 0.2625 = 7.5 \text{ COILS}$$

$$\text{STEP 6 - (EQ. 19-12)} \quad C = \left[\frac{G D_w}{8 K N_a} \right]^{1/3} = \left[\frac{(11.2 \times 10^6)(0.2625)}{8(80)(6.0)} \right]^{1/3} = 9.15 \quad \text{USE } N_a = 6.0 \quad \text{STEP 5}$$

$$K = 1.16 \quad (\text{FIG. 7-12})$$

$$\text{(EQ. 19-4)} \quad T_o = \left(\frac{8 K C F_o}{\pi D_w^2} \right) = \frac{2.546(1.16)(9.15)(220)}{(0.2625)^2} = 86,262 \text{ PSI} \quad \left\{ \begin{array}{l} \text{TDO} \\ \text{HIGH} \end{array} \right\}$$

REPEAT FROM STEP 3: TRY $D_w = 0.2830 \text{ IN}$ U.S. GAGE 1; $T_d = 79,000 \text{ PSI}$

(CONTINUED - NEXT PAGE)

$$T_{max} = 104,000 \text{ PSI}$$

13. (CONTINUED) STEP 4 - $N_{a, \max} = [2.50 - 2(0.2830)] / 0.2830 = 6.83$ USE $N_a = 6$ COILS

STEP 6 - $C = \left[\frac{(11.2 \times 10^6)(0.2830)}{8(80)(6.0)} \right]^{1/3} = 9.38$; $K = 1.155$ STEP 5

OPERATING STRESS - $T_0 = \frac{(2.546)(1.155)(9.38)(220)}{(0.2830)^2} = 75770 \text{ PSI}$ OK

- SOLID LENGTH & STRESS AT SOLID LENGTH

$L_s = D_w(N_a + 2) = 0.2830(6.0 + 2) = 2.264 \text{ IN}$

$F_s = k(L_f - L_s) = (80 \text{ LB/IN})(5.25 - 2.264) \text{ IN} = 239 \text{ LB}$

STRESS AT SOLID HEIGHT $T_s = T_0(F_s/F_0) = 75770(239/220) = 82272 \text{ PSI}$ OK

FINAL GEOMETRY

- $D_m = C D_w = 9.38(0.2830) = 2.655 \text{ IN}$

$D_o = D_m + D_w = 2.655 + 0.2830 = 2.938 \text{ IN}$

$D_i = D_m - D_w = 2.655 - 0.2830 = 2.372 \text{ IN}$

$CC = (L_o - L_s) / N_a = (2.50 - 2.264) / 6.0 = 0.0393 \text{ IN} > \frac{D_w}{10}$ OK

SUMMARY: $D_w = 0.2830 \text{ IN}$, $L_s = 5.25 \text{ IN}$, $L_o = 2.50 \text{ IN}$, $D_m = 2.655 \text{ IN}$

14. METHOD 2 $L_o = 1.75 \text{ IN}$; $F_o = 22.0 \text{ LB}$; $L_i = 3.00 \text{ IN}$; $F_i = 5.0 \text{ LB}$
ASTM A401; SEVERE SERVICE; $T_s \approx 125 \text{ KSI}$

$k = \frac{F_o - F_i}{L_i - L_o} = \frac{22.0 - 5.0}{3.00 - 1.75} = 13.6 \text{ LB/IN} = k$ $f_o = L_f - L_o = 3.368 - 1.75$
 $f_o = 1.618 \text{ IN}$

$L_s = L_i + F_i/k = 3.00 + 5.0/13.6 = 3.368 \text{ IN}$

$D_w = \sqrt{21.4(F_o)/T_s} = \sqrt{21.4(22)/125000} = 0.061 \text{ IN}$

TRY 16 GA.; $D_w = 0.0625 \text{ IN} \rightarrow T_d = 130 \text{ KSI}$; $T_{max} = 188 \text{ KSI}$

$N_{a, \max} = (L_o - 2D_w) / D_w = [1.75 - 2(0.0625)] / 0.0625 = 26$; TRY $N_a = 20$

$C = \left[\frac{G D_w}{8 k N_a} \right]^{1/3} = \left[\frac{(11.2 \times 10^6)(0.0625)}{8(13.6)(20)} \right]^{1/3} = 6.85 \rightarrow K = 1.225$

$T_0 = \frac{2.546 K C F_o}{D_w^2} = \frac{2.546(1.225)(6.85)(22.0)}{(0.0625)^2} = 120325 \text{ PSI}$ OK

$L_s = D_w(N_a + 2) = 0.0625(22) = 1.375 \text{ IN}$

$F_s = k(L_f - L_s) = 13.6(3.368 - 1.375) = 27.1 \text{ LB}$

$T_s = T_0(F_s/F_o) = 120325 \text{ PSI}(27.1/22.0) = 148245 \text{ PSI}$ OK

$D_m = C D_w = 6.85(0.0625) = 0.428 \text{ IN} = D_m$

$D_o = D_m + D_w = 0.491$; $D_i = D_m - D_w = 0.366 \text{ IN}$

$CC = (L_o - L_s) / N_a = (1.75 - 1.375) / 20 = 0.019 \text{ IN} > \frac{D_w}{10}$ OK

BUCKLING: $L_f/D_m = 3.368/0.428 = 7.87$; $\left(\frac{f_o}{L_s}\right)_{CR} = 0.21$ (FIG. 19-15)

$(f_o)_{CR} = 0.21 L_f = 0.21(3.368) = 0.71 \text{ IN}$; ACTUAL $f_o > (f_o)_{CR}$ - WILL BUCKLE

14. (CONTINUED) AFTER SEVERAL ITERATIONS THIS DESIGN WAS PRODUCED WHICH WILL NOT BUCKLE.

$$D_w = 0.072 \text{ IN}; T_d = 129 \text{ KSI}; T_{\text{MAX}} = 185 \text{ KSI}$$

$$N_{\text{MAX}} = 22.3; \text{ USED } N_u = 12; C = 8.52; D_m = 0.613 \text{ IN.}$$

$$T_o = 108200 \text{ PSI}; L_s = 1.008 \text{ IN.}; F_s = 321 \text{ LB}; T_s = 157.9 \text{ KSI}$$

$$CC = 0.062 \text{ IN (4000)}; \text{ BUCKLING - } L_e/D_m = 5.49; \left(\frac{L_e}{L_e} \right)_{\text{CR}} = 0.53$$

$$(f_o)_{\text{CR}} = 0.53 L_e = 1.78 \text{ IN}; \text{ ACTUAL } f_o = 1.62 \text{ IN OK}$$

15. METHOD 2 $L_o = 1.25 \text{ IN}; F_o = 14.0 \text{ LB}; L_i = 2.00 \text{ IN}; F_i = 1.50 \text{ LB}$

ASTM A313, TYPE 302, AVG. SERVICE; $T_d \approx 100 \text{ KSI}$

$$k = \frac{14 - 1.50}{2.00 - 1.25} = 16.67 \text{ LB/IN}; L_e = L_i + \frac{F_i}{k} = 2.00 + \frac{1.50}{16.67} = 2.09 \text{ IN.}$$

$$D_w = \sqrt{21.4(F_o)/T_d} = \sqrt{21.4(14.0)/100000} = 0.055 \text{ IN.}$$

$$\text{TRY 16 GA.}; D_w = 0.0625 \text{ IN}; T_d = 115 \text{ KSI}; T_m = 128 \text{ KSI}$$

$$N_{\text{MAX}} = (L_o - 2D_w)/D_w = [1.25 - 2(0.0625)]/0.0625 = 18.0 \rightarrow \text{USE } N_u = 16$$

$$C = \left[\frac{G D_w}{8(k) N_u} \right]^{1/3} = \left[\frac{(10 \times 10^6)(0.0625)}{8(16.67)(16)} \right]^{1/3} = 6.64 \text{ THEN } K = 1.23$$

$$T_o = \frac{2.546(1.23)(6.64)(14.0)}{(0.0625)^2} = 74540 \text{ PSI OK}$$

$$L_s = D_w(N_u + 2) = 0.0625(18) = 1.125 \text{ IN.}$$

$$F_s = k(L_e - L_s) = 16.67(2.09 - 1.125) = 16.09 \text{ LB}$$

$$T_s = T_o(F_s/F_o) = 74540(16.09/14.0) = 85650 \text{ PSI OK}$$

$$D_m = C D_w = 6.64(0.0625) = 0.415$$

$$L_e/D_m = 2.09/0.415 = 5.04 \text{ NO BUCKLING}$$

$$CC = (L_o - L_s)/N_u = (1.25 - 1.125)/16 = 0.008 \text{ IN} > D_w/N_u \text{ OK}$$

$$D_o = D_m + D_w = 0.415 + 0.0625 = 0.478 \text{ IN}$$

$$D_i = D_m - D_w = 0.415 - 0.0625 = 0.353 \text{ IN.}$$

16

METHOD 2: $L_o = 4.00 \text{ IN}$, $F_o = 250 \text{ LB}$, $L_i = 10.50 \text{ IN}$, $F_i = 60 \text{ LB}$ ASTM A231: $T_u = 90 \text{ KSI}$ FOR SEVERE SERVICE

$$R = \frac{F_o - F_i}{L_i - L_o} = \frac{250 - 60}{10.50 - 4.00} = 29.23 \text{ LB/IN}; L_c = 10.50 + \frac{60 \text{ LB}}{29.23 \text{ LB/IN}} = 12.55 \text{ IN}$$

$$D_w = \sqrt{21.4 F_o / T_u} = \sqrt{21.4 (250) / 90000} = 0.244 \text{ IN.}$$

TRIALS WITH GAGES 2 AND 1 FAILED T_o TOO HIGHFOR $D_w = 0.3065 \text{ IN}$ (0 GAGE), $T_u = 90000 \text{ PSI}$, $T_{max} = 130 \text{ KSI}$

$$N_{a, max} = (L_o - 2D_w) / D_w = (4.00 - 2(0.3065)) / 0.3065 = 11.05; \text{ TRY } N_a = 10$$

$$C = \left[\frac{6 D_w}{8 K N_a} \right]^{1/3} = \left[\frac{(11.2 \times 10^6)(0.3065)}{8(29.23)(10)} \right]^{1/3} = 11.37; \text{ THEN } K = 1.125$$

$$T_o = \frac{(2.546) K C F_o}{D_w^2} = \frac{(2.546)(1.125)(11.37)(250)}{(0.3065)^2} = 86,680 \text{ PSI OK}$$

$$L_s = D_w (N_a + 2) = (0.3065)(12) = 3.678 \text{ IN}$$

$$F_s = R(L_c - L_s) = (29.23)(12.55 - 3.678) = 259 \text{ LB}$$

$$T_s = T_o (F_s / F_o) = (86,680) (259 / 250) = 89,900 \text{ PSI} < T_{max} \text{ OK}$$

$$D_m = C D_w = (11.37)(0.3065) = 3.485 \text{ IN}$$

$$L_s / D_m = 12.55 / 3.485 = 3.60 \text{ NO BUCKLING}$$

$$CC = (L_o - L_s) / N_a = (4.00 - 3.678) / 10 = 0.032 \text{ IN} > D_w / 10 \text{ OK}$$

17

METHOD 2 $F_o = 14.0 \text{ LB}$; $L_o = 0.68 \text{ IN}$; $F_i = 0$; $L_i = L_c = 1.75 \text{ IN}$; $R = \frac{14}{1.75 - 0.68} = 13.08 \text{ LB/IN.}$ MUSIC WIRE; AV. SERV.; $T_u = 120 \text{ KSI}$

$$D_w = \sqrt{21.4 (14) / 120000} = 0.050 \text{ IN}; \text{ USE } D_w = 0.055 \text{ IN, 24 GAGE} \begin{cases} T_u = 135 \text{ KSI} \\ T_n = 150 \text{ KSI} \end{cases}$$

$$N_{a, max} = (L_o - 2D_w) / D_w = (0.68 - 2(0.055)) / 0.055 = 8.9 \text{ CONS}$$

USE $N_a = 8.0$

$$C = \left[\frac{(11.85 \times 10^6)(0.055)}{8(13.08)(8.0)} \right]^{1/3} = 9.20 \rightarrow K = 1.16$$

$$T_o = \frac{2.546 (1.16) (9.20) (14)}{(0.055)^2} = 125,750 \text{ PSI OK}$$

$$L_s = D_w (N_a + 2) = 0.055 (10) = 0.550 \text{ IN}$$

$$F_s = R(L_c - L_s) = 13.08 (1.75 - 0.55) = 15.70 \text{ LB}$$

$$T_s = T_o (F_s / F_o) = 125,750 (15.7 / 14.0) = 141,010 \text{ PSI OK}$$

$$D_m = C D_w = 9.20 (0.055) = 0.506 \text{ IN}$$

$$L_s / D_m = 1.75 / 0.506 = 3.46 \text{ NO BUCKLING}$$

$$CC = (0.68 - 0.55) / 8 = 0.016 \text{ IN} > D_w / 10 \text{ OK}$$

$$D_o = D_m + D_w = 0.561 \text{ IN.}$$

$$D_i = D_m - D_w = 0.451 \text{ IN.}$$

$$k = \frac{8.00}{1.75} = 4.57 \text{ LB/IN}$$

18.

METHOD 2 $L_f = 2.75$; $f_o = 1.75 \text{ IN}$; $L_o = L_f - f_o = 1.00 \text{ IN}$; $F_o = 8.00 \text{ IN}$.

ASTM A313, TYPE 316, AVG. SERV.; $T_d \approx 90 \text{ KSI}$

$$D_{wr} = \sqrt{21.4(8.0)/90,000} = 0.044 \text{ IN} \rightarrow \text{USE } D_{wr} = 0.0475 \text{ IN}; \text{ RGA GE}$$

$$T_d = 0.85(122 \text{ KSI}) = 104 \text{ KSI}; T_M = 0.85(135) = 115 \text{ KSI}$$

$$N_{a, \max} = (L_o - 2D_{wr})/D_{wr} = [1.00 - 2(0.0475)]/0.0475 = 19.05 \rightarrow \text{USE } N_a = 17$$

$$C = \left[\frac{(10.0 \times 10^6)(0.0475)}{8(4.57)(17)} \right]^{1/3} = 9.14 \rightarrow R = 1.16$$

$$T_o = \frac{2.546(1.16)(9.14)(8.0)}{(0.0475)^2} = 95,740 \text{ PSI OK}$$

$$L_s = 0.0475(17+2) = 0.9025 \text{ IN}; F_s = 4.57 \text{ LB/IN}(2.75 - 0.9025) = 8.44 \text{ LB}$$

$$T_s = T_o(F_s/F_o) = 95,740(8.44/8.0) = 101,050 \text{ PSI OK}$$

$$D_m = C D_{wr} = 9.14(0.0475) = 0.434 \text{ IN.}; L_f/D_m = 2.75/0.434 = 6.33 \text{ (HIGH)}$$

$$(f_o/L_f)_{CR} = 0.32; (f_o)_{CR} = 0.32(2.75) = 0.88 \text{ IN}; f_o = 1.75 > 0.88 \text{ (WILL BUCKLE)}$$

19.

SAME AS 18 EXCEPT $D_{rod} = 0.625 \text{ IN}$ - USE METHOD 1

LET $D_m = 0.75 \text{ IN}$; $T_d \approx 100 \text{ KSI}$

$$D_{wr} = \left[\frac{3.06(F_o)(D_m)}{T_d} \right]^{1/3} = \left[\frac{3.06(8.0)(0.75)}{100,000} \right]^{1/3} = 0.057 \rightarrow \text{USE } D_{wr} = 0.0625 \text{ IN.}$$

$$T_d = 0.85(115 \text{ KSI}) = 97.8 \text{ KSI}; T_M = 0.85(128) = 109 \text{ KSI}$$

$$C = D_m/D_{wr} = 0.75/0.0625 = 12.0 \rightarrow R = 1.12$$

$$T_o = \frac{2.546(1.12)(12.0)(8.0)}{(0.0625)^2} = 70,100 \text{ PSI OK}$$

$$N_a = \frac{G D_{wr}}{8 k C^3} = \frac{(10 \times 10^6)(0.0625)}{8(4.57)(12)^3} = 9.89 \text{ COILS}$$

$$L_s = D_{wr}(N_a + 2) = 0.0625(11.89) = 0.743 \text{ IN}$$

$$F_s = k(L_f - L_s) = 4.57(2.75 - 0.743) = 9.17 \text{ LB}$$

$$T_s = T_o(F_s/F_o) = 70,100(9.17/8.0) = 80,400 \text{ PSI OK}$$

$$L_f/D_m = 2.75/0.75 = 3.67 \text{ OK NO BUCKLING}$$

$$CC = (L_o - L_s)/N_a = (1.00 - 0.743)/9.89 = 0.026 \text{ IN GOOD}$$

$$D_i = D_m + D_{wr} = 0.75 + 0.0625 = 0.8125 \text{ IN.}$$

$$D_i - D_m - D_{wr} = 0.75 - 0.0625 = 0.6875 \text{ IN} > D_{rod} \text{ OK}$$

20.

SAME AS 17 EXCEPT $D_{HOLE} = 0.75$ IN. METHOD 1LET $D_m = 0.625$ IN; $T_d = 120$ KSI

$$D_w = \left[\frac{3.06(14.0)(0.625)}{120000} \right]^{1/3} = 0.061 \text{ IN.} \rightarrow \text{USE } D_w = 0.063 \text{ IN. } T_d = 130 \text{ KSI } T_M = 145 \text{ KSI}$$

$$C = D_m/D_w = 0.625/0.063 = 9.92 \rightarrow K = 1.15$$

$$T_o = \frac{2.546(1.15)(9.92)(14.0)}{(0.063)^2} = 102500 \text{ PSI OK}$$

$$N_a = \frac{(1.85 \times 10^6)(0.063)}{8(13.08)(9.92)^3} = 7.31 \text{ COILS; } L_c = 0.063(7.31+2) = 0.586 \text{ IN}$$

$$F_s = k(L - L_s) = 13.08(1.75 - 0.586) = 15.22 \text{ LB; } T_s = T_o F_s/F_o = 111,500 \text{ PSI OK}$$

$$L_f/D_m = 1.75/0.625 = 2.80 \text{ OK NO BUCKLING}$$

$$CC = (0.68 - 0.586)/7.31 = 0.013 \text{ IN} > D_w/10 \text{ OK}$$

$$D_o = 0.625 + 0.063 = 0.688 \text{ IN}$$

$$\text{CLEARANCE WITH HOLE} = 0.75 - 0.688 = 0.062 \text{ IN OK} > D_w/10$$

21.

METHOD 2 $L_i = 3.05$ IN; $F_o = 45$ LB; $L_f = 3.50$ IN; $F_i = 22.0$ LB

$$k = \frac{45 - 22}{3.50 - 3.05} = 57.1 \text{ LB/IN; } L_f = L_i + F_i/k = 3.50 + 22/57.1 = 3.93 \text{ IN}$$

$$D_w = \sqrt{21.4(45)/90000} = 0.103 \text{ IN.} \rightarrow \text{USE } D_w = 0.1055 \text{ IN.}$$

ASTM A231; $T_d = 109$ KSI; $T_M = 157$ KSI; SEVERE SERVICE

$$N_{a, \text{MAX}} = (L_i - 2D_w)/D_w = [3.05 - 2(0.1055)]/0.1055 = 26.9 \text{ COILS} \rightarrow \text{USE } N_a = 18$$

$$C = \left[\frac{G D_w}{8 k N_a} \right]^{1/3} = \left[\frac{(11.2 \times 10^6)(0.1055)}{8(57.1)(18)} \right]^{1/3} = 5.44 \rightarrow K = 1.285$$

$$T_o = \frac{2.546(1.285)(5.44)(45)}{(0.1055)^2} = 71950 \text{ PSI; } < T_d \text{ OK}$$

$$L_c = D_w(N_a + 2) = 0.1055(20) = 2.11 \text{ IN; } F_s = 57.1 \text{ LB/IN}(3.93 - 2.11) = 93.0 \text{ LB}$$

$$T_s = T_o (F_s/F_o) = 71950 (93.0/45.0) = 148700 \text{ PSI} < T_{\text{MAX}} \text{ OK}$$

$$D_m = C D_w = 5.44(0.1055) = 0.574 \text{ IN}$$

$$L_f/D_m = 3.93/0.574 = 6.85; (F_o/L_f)_{cr} = 0.27$$

$$(F_o)_{cr} = 0.27(3.93) = 1.061 \text{ IN.}$$

$$(F_o)_{\text{ACT}} = L_f - L_o = 3.93 - 3.05 = 0.88 \text{ IN} < (F_o)_{cr} \text{ OK NO BUCKLING}$$

$$CC = (3.05 - 2.11)/18 = 0.052 \text{ IN} > \frac{D_w}{10} \text{ OK}$$

$$D_o = D_m + D_w = 0.680 \text{ IN.}$$

$$D_i = D_m - D_w = 0.469 \text{ IN.}$$

22

COMPRESSION SPRING: METHOD 1

ASTM A227 STEEL WIRE; $G = 11.5 \times 10^6 \text{ PSI}$ (TABLE 18-4)
STRESSES FROM FIG. 18-8.

$$L_i = 2.50 \text{ IN}, F_i = 20.0 \text{ LB}, L_o = 2.10 \text{ IN}, F_o = 35 \text{ LB.}$$

INSTALL SPRING AROUND 1.50 IN DIA. SHAFT.

$$\text{SPRING SCALE} = k = \frac{F_o - F_i}{L_i - L_o} = \frac{35 - 20}{2.50 - 2.10} = 37.5 \text{ LB/IN}$$

$$\text{FREE LENGTH} = L_f = L_i + F_i/k = 2.50 + 20.0/37.5 = 3.033 \text{ IN.}$$

$$\text{TRY } D_m = 1.75 \text{ IN.}; D_{w_{\max}} = D_m - D_{sh} = 1.75 - 1.50 = 0.25 \text{ IN.}$$

$$\text{SEVERE SERVICE: } T_d \approx 85,000 \text{ PSI}$$

$$D_w \approx \left[\frac{(3.06)(F_o)(D_m)}{T_d} \right]^{1/3} = \left[\frac{(3.06)(35)(1.75)}{85,000} \right]^{1/3} = 0.130 \text{ IN.}$$

$$\text{TRY U.S. WIRE GA. 9: } D_w = 0.1483 \text{ IN.}$$

$$T_d = 87,000 \text{ PSI (SEVERE SERV.)}; T_{\max} = 116,000 \text{ PSI (LT. SERV.)}$$

$$C = D_m/D_w = 1.75/0.1483 = 11.80; K = 1.125 \text{ (FIG. 18-14)}$$

$$T_o = \frac{8KF_oD_m}{\pi D_w^3} = \frac{8(1.125)(35)(1.75)}{\pi (0.1483)^3} = 53,800 \text{ PSI OK}$$

$$N_a = \frac{G D_w}{8k C^3} = \frac{(11.5 \times 10^6)(0.1483)}{8(37.5)(11.80)^3} = 3.45 \text{ COILS}$$

$$L_s = D_w(N_a + 2) = (0.1483)(3.45 + 2) = 0.809 \text{ IN.}$$

$$F_s = k(L_f - L_s) = (37.5)(3.033 - 0.809) = 83.4 \text{ LB}$$

$$T_s = T_o \cdot F_s/F_o = (53,800 \text{ PSI}) \cdot (83.4/35) = 128,200 \text{ PSI TOO HIGH}$$

$$\text{TRY } D_w = 0.162 \text{ IN. - 8 GAUGE: } T_d = 86,000 \text{ PSI}; T_{\max} = 114,000 \text{ PSI}$$

SUMMARY OF RESULTS:

$$C = 10.80; K = 1.13$$

$$T_o = 41,455 \text{ PSI OK}$$

$$N_a = 4.93 \text{ COILS}; L_s = 1.122 \text{ IN}; F_s = 71.7 \text{ LB}$$

$$T_s = 84,900 \text{ PSI OK}$$

$$ID = D_m - D_w = 1.75 - 0.162 = 1.588 \text{ IN OK}$$

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COMPRESSION SPRING: METHOD 1: METRICASTM A227 STEEL WIRE: $G = 79.3 \text{ GPa} = 79.3 \times 10^3 \text{ MPa}$ (TABLE 18-4) $L_i = 60 \text{ mm}$, $F_i = 90 \text{ N}$, $L_o = 50 \text{ mm}$, $F_o = 155 \text{ N}$

$$k = \frac{F_o - F_i}{L_i - L_o} = \frac{155 - 90}{60 - 50} = 6.5 \text{ N/mm}$$

$$L_s = L_i + F_i/k = 60 \text{ mm} + 90 \text{ N} / 6.5 \text{ N/mm} = 73.85 \text{ mm}$$

SPRING INSTALLED ON 38 mm DIA. SHAFT: TRY $D_m = 45 \text{ mm}$ SEVERE SERVICE: $T_s \approx 550 \text{ MPa}$ (APP. A191)

$$\text{TRIAL } D_w = \left[\frac{(3.06)(F_o)(D_m)}{T_s} \right]^{1/3} = \left[\frac{(3.06)(155)(45)}{550} \right]^{1/3} = 3.39 \text{ mm}$$

TRY $D_w = 3.80 \text{ mm}$ (TABLE 18-2); $T_s = 600 \text{ MPa}$, $T_{\max} = 800 \text{ MPa}$

$$C = D_m / D_w = 45 / 3.80 = 11.84; K = 1.125 \text{ (FIG. 18-14)}$$

$$T_o = \frac{8 K F_o D_m}{\pi D_w^3} = \frac{8(1.125)(155)(45)}{\pi (3.80)^3} = 364 \text{ MPa OK}$$

$$N_a = \frac{G D_w}{8 K C^3} = \frac{(79.3 \times 10^3)(3.80)}{8(6.5)(11.84)^3} = 3.49 \text{ COILS}$$

$$L_s = D_w(N_a + 2) = (3.80)(3.49 + 2) = 20.86 \text{ mm}$$

$$F_s = k(L_s - L_o) = (6.5 \text{ N/mm})(73.85 - 20.86) \text{ mm} = 344 \text{ N}$$

$$T_s = T_o \frac{F_s}{F_o} = 364 \text{ MPa} \left(\frac{344 \text{ N}}{155 \text{ N}} \right) = 809 \text{ MPa (JUST SLIGHTLY HIGH)}$$

$$ID = D_m - D_w = 45 - 3.80 = 41.2 \text{ mm OK}$$

24

COMPRESSION SPRING: ANALYSIS: ASTM A229, $G = 11.2 \times 10^6 \text{ PSI}$ 17 GA.: $D_w = 0.054 \text{ IN.}$; $T_s = 106 \text{ KSI (SEVERE)}$; $T_o = 128 \text{ KSI (AVG)}$; $T_s = 141 \text{ KSI (LT.)}$ OD = 0.531 IN: $D_m = OD - D_w = 0.531 - 0.054 = 0.477 \text{ IN.}$

$$C = D_m / D_w = 0.477 / 0.054 = 8.83; K = 1.17; N = 7.0 \text{ COILS}; N_a = 7.0 - 2 = 5.0$$

FOR $F_o = 10.0 \text{ LB.}$:

$$T_o = \frac{8 K F_o C}{\pi D_w^3} = \frac{8(1.17)(10)(8.83)}{\pi (0.054)^3} = 90250 \text{ PSI OK FOR SEVERE SERV.}$$

$$f_o = \frac{8(F_o)(C^3)(N_a)}{G D_w} = \frac{8(10)(8.83)^3(5.0)}{(11.2 \times 10^6)(0.054)} = 0.456 \text{ IN.}$$

$$L_o = L_f - f_o = 1.25 \text{ IN} - 0.456 = 0.794 \text{ IN.}; L_s = D_w(N) = 0.054(7) = 0.378 \text{ IN.}$$

$$k = \frac{F_o}{L_f - L_o} = \frac{10.0 \text{ LB.}}{(1.25 - 0.794) \text{ IN.}} = 21.93 \text{ LB./IN.}$$

$$F_s = k(L_f - L_s) = (21.93)(1.25 - 0.378) = 19.12 \text{ LB.}$$

$$T_s = T_o \frac{F_s}{F_o} = (90250 \text{ PSI}) \frac{19.12 \text{ LB.}}{10.0 \text{ LB.}} = 172,600 \text{ PSI TOO HIGH}$$

BUT $T_{\max} \approx T_s$ FOR LIGHT SERVICE = 141,000 PSI

25

EXTENSION SPRING $F_o = 7.75 \text{ LB}$; $L_o = 2.75 \text{ IN}$; $L_i = 2.25 \text{ IN}$, $F_i = 5.25 \text{ LB}$
 $D_o < 3.00 \text{ IN}$; MUSIC WIRE, SEVERE SERVICE. INSTALLED

LET $D_m = 0.250 \text{ IN}$; $T_u \approx 120,000 \text{ PSI}$

$$D_w = \left[\frac{8 K F_o D_m}{\pi T_u} \right]^{2/3} = \left[\frac{8 (1.20) (7.75) (0.250)}{\pi (120,000)} \right]^{2/3} = 0.0367 \text{ IN}$$

USE 16 GA., $D_w = 0.037 \text{ IN}$; $T_u = 120,000 \text{ PSI}$

$$D_o = D_m + D_w = 0.287 \text{ IN} \text{ OK}; D_i = D_m - D_w = 0.213 \text{ IN.}$$

$$C = D_m / D_w = 0.25 / 0.037 = 6.76 \rightarrow K = 1.225$$

$$T_o = \frac{8 K F_o D_m}{\pi D_w^3} = \frac{8 K F_o C}{\pi D_w^2} = \frac{8 (1.225) (7.75) (6.76)}{\pi (0.037)^2} = 119,320 \text{ PSI OK}$$

$$k = \frac{\Delta F}{\Delta L} = \frac{7.75 - 5.25}{2.75 - 2.25} = 5.0 \text{ LB/IN.}$$

$$N_a = \frac{G D_w}{8 C^3 k} = \frac{(11.85 \times 10^6) (0.037)}{8 (6.76)^3 (5.0)} = 35.5 \text{ COILS}$$

$$\text{BODY LENGTH} \approx D_w (N_a + 1) = 0.037 (36.5) = 1.350 \text{ IN.} = \text{B.L.}$$

ASSUME FULL LOOP AT EACH END

$$L_f = \text{B.L.} + 2 D_i = 1.350 + 2 (0.213) = 1.777 \text{ IN.}$$

$$\text{DEFLECTION TO } L_o: f_o = L_o - L_f = 2.750 - 1.777 = 0.973 \text{ IN}$$

$$\text{INITIAL FORCE} = F_i = F_o - k f_o = 7.75 - 5.0 (0.973) = 2.88 \text{ LB}$$

$$\text{INITIAL STRESS} = T_i = T_o \times \frac{F_i}{F_o} = 119,320 \times \frac{2.88}{7.75} = 44,340 \text{ PSI}$$

FROM FIG 19-21 - STRESS SHAPOLE $\approx 14-21 \text{ KSI}$ HIGH

USE SMALLER END LOOPS - SAY 0.06 EACH

$$L_f = \text{B.L.} + 2 (0.06) = 1.35 + 0.12 = 1.47 \text{ IN}$$

$$f_o = L_o - L_f = 2.750 - 1.47 = 1.28 \text{ IN.}$$

$$F_i = 7.75 - 5.0 (1.28) = 1.35 \text{ LB}$$

$$T_i = T_o \times \frac{F_i}{F_o} = 119,320 \times \frac{1.35}{7.75} = 20,800 \text{ PSI OK}$$

SUMMARY:

$D_w = 0.037 \text{ IN}$ - 16 GA. MUSIC WIRE

$D_m = 0.250 \text{ IN}$; $D_o = 0.287 \text{ IN}$; $D_i = 0.213 \text{ IN.}$

35.5 COILS

2 LOOPS @ 0.06 IN. EACH

$F_o = 7.75 \text{ LB}$ @ 2.75 IN - OPERATING

$F_i = 5.25 \text{ LB}$ @ 2.25 IN - INSTALLED

$F_i = 1.35 \text{ LB}$ @ 1.47 IN - INITIAL

26 EXTENSION: MUSIC WIRE, AVG. SERVICE, $T_u \approx 130 \text{ ksi}$
 $F_0 = 15.0 \text{ LB}$; $F_L = 5.20 \text{ LB}$; $L_0 = 5.00 \text{ in}$; $L_A = 3.75 \text{ in}$.
 $R = 7.84 \text{ LB/in}$; $D_m = 0.60 \text{ in}$; $D_{m_{\text{TRIAL}}} = 0.0596 \text{ in}$; $D_m = 0.063 (26 \text{ Ga.})$
 $T_u = 132,000 \text{ PSI}$; $C = 9.52$; $K = 1.153$; $T_0 = 105,600 \text{ PSI OK}$; $N_A = 13,78$
 BODY LENGTH = B.L. = 0.93 in ; FOR ENO LOOP SIZE = $0.537 \text{ in} (= ID)$; $F_I = -8.48 \text{ LB}$
 CHANGE E.L.S. = $1.20 \text{ in} \rightarrow F_I = 1.92 \text{ LB}$; $T_I = 13500 \text{ PSI (OK FIG. 18-21)}$
 $OD = 0.663 \text{ in} < 0.75 \text{ in. OK}$

27 EXTENSION: MUSIC WIRE, SEVERE SERVICE; $T_u \approx 100 \text{ ksi}$; $F_0 = 11.0 \text{ LB}$; $L_0 = 3.00 \text{ in}$,
 $R = 6.80 \text{ LB/in}$; $D_m = 0.65 \text{ in}$; $D_{m_{\text{TRIAL}}} = 0.058 \text{ in} \rightarrow \text{USED } D_m = 0.059 \text{ in} (25 \text{ Ga.})$
 $T_u = 110 \text{ ksi}$; $OD = 0.709 \text{ in}$; $ID = 0.591$; $C = 11.02$; $K = 1.131$; $T_0 = 91,125 \text{ PSI OK}$
 $N_A = 9.61 \text{ COILS}$; B.L. = 0.626 in ; E.L.S. = $0.591 \text{ in} = ID$; $F_I = 1.89 \text{ LB}$; $T_I = 17.3 \text{ ksi HIGH}$
 CHANGE E.L.S. = 0.54 in ; $F_I = 1.20 \text{ LB}$; $T_I = 10.95 \text{ ksi (OK FIG. 18-21)}$

28 EXTENSION: MUSIC WIRE, SEVERE SERV.; $F_0 = 10.0 \text{ LB}$; $L_0 = 6.10 \text{ in}$; $T_u = 100 \text{ ksi}$
 (SAME AS #24 FOR D_m , D_m , OD , ID , C , K , T_0); $N_A = 25.14 \text{ COILS}$; B.L. = 1.542 in
 E.L.S. = $0.591 \text{ in} = ID$; $F_I = 1.48 \text{ LB}$; $T_I = 13500 \text{ PSI (OK FIG. 18-21)}$

29 EXTENSION: MUSIC WIRE; AVERAGE SERV.; $F_0 = 10.0 \text{ LB}$; $L_0 = 9.4 \text{ in}$; $R = 1.50 \text{ LB/in}$.
 $T_u \approx 130 \text{ ksi}$; $D_m = 0.6875 \text{ in. (11/16)}$; $D_{m_{\text{TRIAL}}} = 0.0545 \text{ in}$; $\text{USED } D_m = 0.055 \text{ in}$,
 $T_u = 135 \text{ ksi}$; (24 Ga.); $OD = 0.7425 \text{ in}$; $ID = 0.6325 \text{ in}$; $C = 12.5$; $K = 1.114$
 $T_0 = 117,300 \text{ PSI}$; $N_A = 27.81 \text{ COILS}$; B.L. = 1.58 in ; E.L.S. = $0.6325 \text{ in} = ID \rightarrow$
 $F_I = -0.14 \text{ LB}$; CHANGE E.L.S. = 1.0 in ; $F_I = 0.96 \text{ LB}$; $T_I = 113 \text{ ksi (OK)}$

30 EXTENSION: ASTM A313; TYPE 302; $F_0 = 162 \text{ LB}$; $L_0 = 10.80 \text{ in}$; $R = 38.0 \text{ LB/in}$
 $T_u \approx 80 \text{ ksi (AVG. SERV.)}$; $D_m = 1.50 \text{ in}$; $D_{m_{\text{TRIAL}}} = 0.210 \text{ in} \rightarrow D_m = 0.2253 \text{ in (4 Ga.)}$
 $T_u = 81 \text{ ksi}$; $OD = 1.7253 \text{ in}$; $ID = 1.2747 \text{ in}$; $C = 6.66$; $K = 1.225$
 $T_0 = 66,312 \text{ PSI}$; $N_A = 25.11 \text{ COILS}$; B.L. = 5.88 in ; E.L.S. = $1.27 \text{ in} = ID$,
 $F_I = 71.7 \text{ LB}$; $T_I = 29,300 \text{ PSI (TIN HIGH)}$.
 CHANGE E.L.S. = 0.90 in ; $F_I = 43.6 \text{ LB}$; $T_I = 17.8 \text{ ksi (OK FIG. 18-21)}$

31

EXTENSION SPRING ENDS

19 Ga.: $D_m = 0.041$ in; $D_m = 0.28$ in; $R_1 = 0.25$ in; $R_2 = 0.094$ in

$\sigma_d = 146$ ksi; $\tau_d = 130$ ksi FOR AVERAGE SERVICE

BENDING (EQ. 19-13) (FIG. 18-22)

$$C_1 = 2R_1/D_m = 2(.25)/.041 = 12.20$$

$$K_1 = \frac{4C_1^2 - C_1 - 1}{4C_1(C_1 - 1)} = 1.065$$

$$\sigma_A = \frac{16(.28)(5.0)(1.065)}{\pi(.041)^3} + \frac{4(5.0)}{\pi(.041)^2} = 114,000 \text{ PSI } \underline{OK}$$

TORSION (EQ. 18-15)

$$C_2 = 2R_2/D_m = 2(.094)/.041 = 4.59$$

$$K_2 = \frac{4C_2 - 1}{4C_2 - 4} = 1.209$$

$$\tau_B = \frac{8D_m F_0 K_2}{\pi(D_m)^3} = \frac{8(.28)(5.0)(1.209)}{\pi(.041)^3} = 62,600 \text{ PSI } \underline{OK}$$

32

TORSION SPRING: ASTM A313, TYPE 302, AVG. SERVICE

ASSUME $D_m = 0.420$ in; $\sigma_d = 140,000$ PSI; $K_b = 1.15$; $\theta_t = 180^\circ = 0.50$ REV.

$$D_m = \left[\frac{32 M_0 K_b}{\pi \sigma_d} \right]^{1/3} = \left[\frac{32(1.0)(1.15)}{\pi(140,000)} \right]^{1/3} = 0.0437 \text{ in } \left\{ \begin{array}{l} \text{USE } 18 \text{ Ga. } D_m = 0.0475 \text{ in} \\ \sigma_d = 160,000 \text{ PSI} \end{array} \right.$$

$$C = D_m/D_w = 0.420/0.0475 = 8.84; k = \frac{4C^2 - C - 1}{4C(C - 1)} = 1.092$$

$$\text{ACTUAL } \sigma = \frac{32 M_0 K_b}{\pi D_m^3} = \frac{32(1.0)(1.092)}{\pi(0.0475)^3} = 103,800 \text{ PSI } \underline{OK}$$

$$k_0 = 1.0 \text{ LB-IN} / 0.50 \text{ REV} = 2.00 \text{ LB-IN/REV}$$

$$N_a = \frac{E D_w^4}{10.2 D_m (k_0)} = \frac{(28 \times 10^6)(0.0475)^4}{10.2(0.420)(2.00)} = 16.64 \text{ COILS TOTAL}$$

ENDS: SPECIFY $L_1 = L_2 = 0.75$ in

$$N_e = \frac{L_1 + L_2}{3\pi D_m} = \frac{0.75 + 0.75}{3\pi(0.420)} = 0.38 \text{ COIL (ENDS)}$$

$$N_b = N_a - N_e = 16.64 - 0.38 = 16.26 \text{ COILS IN BODY}$$

$$\text{OPERATING } D_m = \frac{D_m \pm N_a}{N_a + \theta_t} = \frac{0.420(16.64)}{(16.64 + 0.50)} = 0.408 \text{ in}$$

$$\text{MINIMUM ID} = 0.408 \text{ in} - D_w = 0.408 - 0.0475 = 0.360 \text{ in}$$

$$\text{MAKE ROD } 90\% \text{ OF ID}_{\min} \approx 0.9(0.360) = 0.324 \text{ in}$$

USE $D_{RM} = 0.300$ in (STD. SIZE)

$$L = D_m(N_a + 1 + \theta_t) = 0.0475(16.64 + 1 + 0.5) = 0.862 \text{ in IF COILS TOUCH.}$$

$$D_o = D_m + D_w = 0.420 \text{ in} + 0.0475 = 0.4675 \text{ in. } \underline{OK}$$

33

TORSION SPRING: ASTM A313, TYPE 302, SEVERE SERVICE

ASSUME $D_m = 1.125$; $\sigma_d = 110 \text{ KSI}$; $K_b = 1.15$; $\theta_c = 270^\circ = 0.75 \text{ REV.}$

$$D_{mr} = \left[\frac{32 M_o K_b}{\pi \sigma_d} \right]^{1/3} = \left[\frac{32 (12.0) (1.15)}{\pi (110,000)} \right]^{1/3} = 0.1085 \text{ IN} \left\{ \begin{array}{l} \text{USE 11 GA.; } D_m = 0.1205 \text{ IN} \\ \sigma_d = 118,000 \text{ PSI} \end{array} \right.$$

$$C = D_m / D_{mr} = 1.125 / 0.1205 = 9.34; K_b = \frac{4C^2 - C - 1}{4C(C-1)} = 1.087$$

$$\sigma = \frac{32 M_o K_b}{\pi D_{mr}^3} = \frac{32 (12.0) (1.087)}{\pi (0.1205)^3} = 75,920 \text{ PSI OK}$$

$$R_o = M_o / \theta_c = 12.0 \text{ LB-IN} / 0.75 \text{ REV} = 16.0 \text{ LB-IN/REV}$$

$$N_a = \frac{E D^4}{10.2 D_m (R_o)} = \frac{(29 \times 10^6) (0.1205)^4}{10.2 (1.125) (16.0)} = 32.15 \text{ COILS TOTAL}$$

$$\text{ENDS: LET } L_1 = L_2 = 1.50 \text{ IN; } N_e = \frac{L_1 + L_2}{3\pi (D_m)} = 0.28 \text{ COIL (ENDS)}$$

$$\text{BODY } N_b = N_a - N_e = 32.15 - 0.28 = 31.87 \text{ COILS}$$

$$\text{OPERATING } D_m = \frac{D_m \pm N_a}{N_a + \theta_c} = \frac{(1.125) (32.15)}{(32.15 + 0.75)} = 1.100 \text{ IN}$$

$$\text{I.D.}_{\text{MIN}} = 1.100 - 0.1205 = 0.9795 \text{ IN; } D_{R00} \approx 0.9 (0.9795) = 0.882 \text{ IN}$$

$$\text{USE } D_{R00} = 7/8 \text{ IN} = 0.875 \text{ IN; } L = D_m (N_a + 1 + \theta_c) = 0.1205 (32.15 + 1.75)$$

$$L = 4.08 \text{ IN; OD} = 1.125 + 0.1205 = 1.246 \text{ IN}$$

34

TORSION SPRING: MUSIC WIRE; SEVERE SERVICE; $\theta_c = 1.0 \text{ REV}$ $D_m = 0.625 \text{ IN.}; \sigma_d \approx 140 \text{ KSI}; K_b \approx 1.15$

$$D_{mr} = \left[\frac{32 M_o K_b}{\pi \sigma_d} \right]^{1/3} = \left[\frac{32 (2.50) (1.15)}{\pi (140,000)} \right]^{1/3} = 0.10594 \text{ IN} \left\{ \begin{array}{l} \text{USE 26 GA.; } D_m = 0.063 \\ \sigma_d = 156,000 \text{ PSI} \end{array} \right.$$

$$C = D_m / D_{mr} = 0.625 / 0.063 = 9.92; K_b = \frac{4C^2 - C - 1}{4C(C-1)} = 1.081$$

$$\sigma = \frac{32 (2.50) (1.081)}{\pi (0.063)^3} = 110,100 \text{ PSI OK}$$

$$R_o = 2.50 \text{ LB-IN/REV.}; N_a = \frac{(29 \times 10^6) (0.063)^4}{10.2 (0.625) (2.50)} = 28.66 \text{ COILS}$$

SPECIFY ENDS: $L_1 = L_2 = 1.50 \text{ IN.}$

$$N_e = \frac{2(1.5)}{3\pi (0.125)} = 0.51 \text{ COIL; } N_b = 28.66 - 0.51 = 28.15 \text{ COILS}$$

$$\text{MINIMUM } D_m = \frac{D_m \pm N_a}{N_a + \theta_c} = \frac{0.625 (28.66)}{28.66 + 1.0} = 0.604 \text{ IN}$$

$$\text{MIN I.D.} = 0.604 - 0.063 = 0.541 \text{ IN}$$

$$D_{R00} \approx 0.9 (0.541) = 0.487 \text{ IN} \rightarrow \text{USE } D_R = 7/16 \text{ IN.} = 0.4375 \text{ IN.}$$

$$\text{OD} = 0.625 + 0.063 = 0.688 \text{ IN}$$

$$L = 0.063 (28.66 + 1 + 1.0) = 1.93 \text{ IN}$$

35

TORSION SPRING: $D_w = 0.038$; $D_o = 0.368$ IN; $N_b = 9.50$ COILS
 $L_1 = 0.50$ IN; $L_2 = 1.125$ IN; A401 STEEL; $\theta = 180^\circ = 0.50$ REV

FROM EQ. 7-21, SOLVE FOR M :

$$M_o = \frac{E D_w^4 \theta}{10.2 (D_m) (N_a)} = \frac{(29.5 \times 10^6) (0.038)^4 (0.50)}{10.2 (0.330) (10.02)} = 0.91 \text{ LB-IN}$$

$$\text{WHERE: } D_m = D_o - D_w = 0.368 - 0.038 = 0.330 \text{ IN}$$

$$\text{ENDS: } N_e = \frac{L_1 + L_2}{3\pi (D_m)} = \frac{0.50 + 1.125}{3\pi (0.330)} = 0.52 \text{ COIL}$$

$$N_a = N_b + N_e = 9.50 + 0.52 = 10.02 \text{ COILS}$$

$$\sigma = \frac{32 M_o K_b}{\pi (D_w)^3} = \frac{32 (0.91) (1.094)}{\pi (0.038)^3} = 184,800 \text{ PSI}$$

$$\text{WHERE } C = D_m / D_w = 0.330 / 0.038 = 8.68$$

$$K_b = \frac{4C^2 - C - 1}{4C(C-1)} = 1.094$$

FOR SEVERE SERVICE $\sigma = 200,000 \text{ PSI}$; $\sigma = 184,800 \text{ PSI}$ OK

CHAPTER 19 FASTENERS

- 4 LOAD PER BOLT = $6000 \text{ LB} / 4 = 1500 \text{ LB}$
 SELECT GRADE 2: $\sigma_u = .75 S_{\text{proof}} = .75(55000) = 41250 \text{ PSI}$
[SAE GRADE 2]
TABLE 19-2
 $A_t = \frac{1500}{41250} = 0.0364 \text{ IN}^2$
- USE $1/4$ -28 UNF OR $5/16$ -18 UNC
 FOR $5/16$ -18: $T = KDP = 0.15(.3125)(1500) = 70.3 \text{ LB-IN}$
- 5 $F = A_t S_{\text{pr}} = (0.0140 \text{ IN}^2)(85000 \text{ LB/IN}^2) = 1190 \text{ LB}$
TABLE 19-2
- 6 M4X.5: $A_t = 9.79 \text{ mm}^2$; GRADE 8.6 (SECTION 19-2)
 TENSILE STRENGTH $\approx 800 \text{ MPa}$
 YIELD STRENGTH $\approx 0.6 (T.S.) = .6(800) = 480 \text{ MPa}$
 PROOF STRENGTH $\approx 0.9 (Y.S.) = .9(480) = 432 \text{ MPa}$
 $F = A_t \cdot S_{\text{pr}} = (9.79 \text{ mm}^2)(432 \text{ N/mm}^2) = 4229 \text{ N} = \underline{4.23 \text{ kN}}$
TABLE 19-5
- 7 $7/8$ -14 UNF; $D = 0.875 \text{ IN} \times 25.4 \text{ mm/IN} = 22.2 \text{ mm}$
 $p = 1/m = 1/14 = 0.0714 \text{ IN} \times 25.4 \text{ mm/IN} = 1.81 \text{ mm}$
 NEAREST METRIC THREAD = M24X2 TABLE 19-5
 DIFFERENCE IN D = $24.0 - 22.2 = 1.8 \text{ mm} = .071 \text{ IN}$
 METRIC THREAD IS APPROX. 8% LARGER IN THIS SIZE
- 8 PITCH = $.630/20 = 0.0315 \text{ IN} \times 25.4 \text{ mm/IN} = 0.800 \text{ mm}$
 $D = 0.196 \text{ IN} \times 25.4 \text{ mm/IN} = 4.98 \text{ mm}$
CLOSEST STANDARD THREAD IS M5X0.8
 AMERICAN STANDARD #10-32 IS SIMILAR BUT NOT
 AS CLOSE TO MEASURED DIMENSIONS.
- 9 M10X1.5: $A_t = 58.0 \text{ mm}^2$
 NYLON 6/6: T.S. = 146 MPa ; MAX. $\sigma_u = 0.75(T.S.) = 109.5 \text{ MPa}$
 $F = A_t \sigma_u = (58.0 \text{ mm}^2)(109.5 \text{ N/mm}^2) = 6351 \text{ N} = \underline{6.351 \text{ kN}}$

10

$$\frac{1}{4} - 20 \text{ UNC}; A_t = 0.0318 \text{ in}^2; \sigma_a = 0.5(TS)$$

$$a) \text{SAE GRADE 2: } \sigma_a = 0.5(74) = 37 \text{ ksi}; F = A_t \sigma_a = (0.0318)(77000) \\ F = 1177 \text{ LB}$$

$$b) \text{SAE GRADE 5: } \sigma_a = 0.5(120) = 60 \text{ ksi}; F = 1908 \text{ LB}$$

$$c) \text{SAE GRADE 8: } \sigma_a = 0.5(150) = 75 \text{ ksi}; F = 2385 \text{ LB}$$

$$d) \text{ASTM A367: } \sigma_a = 0.5(60) = 30 \text{ ksi}; F = 954 \text{ LB}$$

$$e) \text{ASTM A574: } \sigma_a = 0.5(180) = 90 \text{ ksi}; F = 2862 \text{ LB}$$

$$f) \text{METRIC GRADE 8.8: } \sigma_a = 0.5(830 \text{ MPa}) = 415 \text{ MPa} \left(\frac{1.0 \text{ ksi}}{6.895 \text{ MPa}} \right) = 60.2 \text{ ksi}$$

$$F = A_t \sigma_a = (0.0318 \text{ in}^2)(60200 \text{ LB/in}^2) = 1914 \text{ LB} \text{ SIMILAR TO SAE GR. 5}$$

$$F = 1914 \text{ LB} \times 4.448 \text{ N/LB} = 8513 \text{ N} = 8.51 \text{ kN}$$

$$g) \text{ALUM, 2024-T4: } \sigma_a = 0.5(68) = 34 \text{ ksi}; F = 1081 \text{ LB}$$

$$h) \text{S43000, ANN.; } \sigma_a = 0.5(75) = 37.5 \text{ ksi}; F = 1193 \text{ LB}$$

$$i) \text{T1-602-4V, ANN.; } \sigma_a = 0.5(130) = 65 \text{ ksi}; F = 2067 \text{ LB}$$

$$j) \text{NYLON 66 DRY: } \sigma_a = 0.5(21) = 10.5 \text{ ksi}; F = 334 \text{ LB}$$

$$k) \text{POLYCARBONATE: } \sigma_a = 0.5(9) = 4.5 \text{ ksi}; F = 143 \text{ LB}$$

$$l) \text{ABS (HIGH IMPACT): } \sigma_a = 0.5(5) = 2.5 \text{ ksi}; F = 80 \text{ LB}$$

ALTERNATE FOR h): SCREW MAY BE FULL HARD - APP. 6

S43000

$$S_u = 90 \text{ ksi}; \sigma_a = 0.5(90) = 45 \text{ ksi}; F = 1431 \text{ LB}$$

CHAPTER 20

MACHINE FRAMES, BOLTED CONNECTIONS, AND WELDED JOINTS

1. DIRECT SHEAR, 5 BOLTS, A307

$$F = \frac{12000 \text{ LB}}{5} = 2400 \text{ LB/BOLT}$$

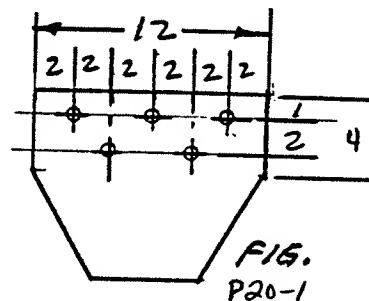
SINGLE SHEAR

$$\tau = F/A_s$$

$$\text{REQ'D } A_s = \frac{F}{\tau_s} = \frac{2400 \text{ LB}}{10000 \text{ LB/IN}^2} = 0.240 \text{ IN}^2$$

$$A_s = \pi D^2/4 : D = \sqrt{4A_s/\pi} = \sqrt{4(0.240)/\pi} = 0.55 \text{ IN}$$

USE 9/16-12 UNC BOLTS, 1 3/4 IN LONG



2. DIRECT SHEAR, EACH BAR SUPPORTS 4 HANGERS

$$F = 4(750) = 3000 \text{ LB} = \text{TOTAL LOAD ON JOINT}$$

4 BOLTS IN DOUBLE SHEAR, A307

$$\text{TOTAL } A_s \text{ REQ'D} = \frac{F}{\tau_s} = \frac{3000 \text{ LB}}{10000 \text{ LB/IN}^2} = 0.30 \text{ IN}^2$$

$$A_s = (2)(4) \frac{\pi D^2}{4} = 2\pi D^2$$

$$D = \sqrt{A_s/(2\pi)} = \sqrt{0.30/(2\pi)} = 0.219 \text{ IN}$$

USE 1/4-20 UNC BOLTS, 2.00 IN LONG

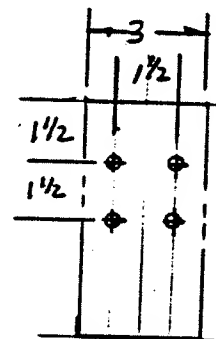


FIG. P20-2

3.

4 BOLTS

HORIZ. DIRECT SHEAR: $F_1 = 5196/4 = 1299 \text{ LB/BOLT}$

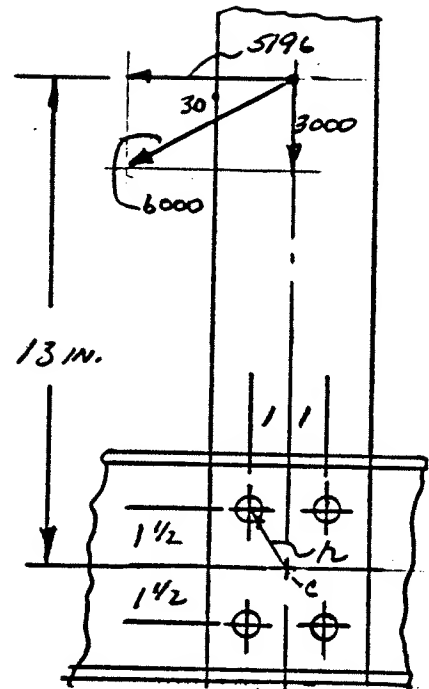
VERTICAL DIR. SHEAR: $F_2 = 3000/4 = 750 \text{ LB} \downarrow$

$M = 5196(13) = 67548 \text{ LB} \cdot \text{IN.}$

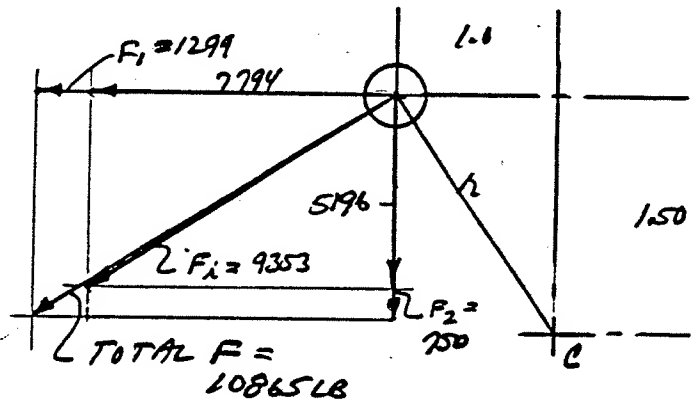
$r = \sqrt{1.50^2 + 1^2} = 1.80 \text{ IN - ALL BOLTS}$

$\Sigma r^2 = 4r^2 = 4(1.80)^2 = 13.0 \text{ IN}^2$

$F_1' = \frac{M r_1'}{\Sigma r^2} = \frac{(67548)(1.80)}{13.0} = 9353 \text{ LB/BOLT}$



FORCES ON BOLT AT UPPER LEFT:



TOTAL $F_x = 7794 + 1299 = 9093$

TOTAL $F_y = 5196 + 750 = 5946$

TOTAL $F = \sqrt{9093^2 + 5946^2}$
 $F = 10865 \text{ LB}$

A 490 BOLTS, DOUBLE SHEAR

$A_s = \frac{F}{T_u} = \frac{10865 \text{ LB}}{22000 \text{ LB/IN}^2} = 0.494 \text{ IN}^2 = 2(\pi D^2/4) = \pi D^2/2$

REQ'D $D = \sqrt{2A_s/\pi} = \sqrt{2(.494)/\pi} = 0.561 \text{ IN.}$

USE 9/16-12 UNC, 3 1/2 IN LONG

4. FIXED-END BEAM - CASE C APPENDIX 14-3.

$$M = \frac{Wl}{8} = \frac{(3000)(87.66)}{8} = 32873 \text{ LB}\cdot\text{IN}$$

$$\text{EACH SIDE: } M = 32873 \text{ LB}\cdot\text{IN.}$$

$$\text{DIRECT SHEAR} = \frac{1500 \text{ LB}}{8 \text{ BOLTS}} = 188 \text{ LB/BOLT}$$

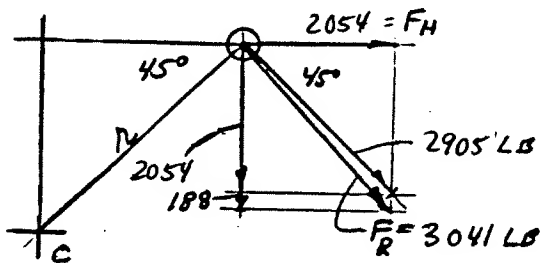
USE 4 BOLTS ON EACH FACE OF COLUMN THROUGH FLANGE AND PLATE.

FORCE ON EACH BOLT DUE TO MOMENT:

$$r = 1.414 \text{ IN}; \Sigma r^2 = 8(1.414)^2 = 16.00 \text{ IN}^2$$

$$F = \frac{Mr}{\Sigma r^2} = \frac{(32873)(1.414)}{16.00} = 2905 \text{ LB/BOLT}$$

FORCES ON BOLT AT UPPER RIGHT:



$$\text{TOTAL } F_V = 2054 + 188 = 2242 \text{ LB}$$

$$F_R = \sqrt{2242^2 + 2054^2} = 3041 \text{ LB}$$

SPECIFY ASTM A 325 HIGH STRENGTH BOLTS

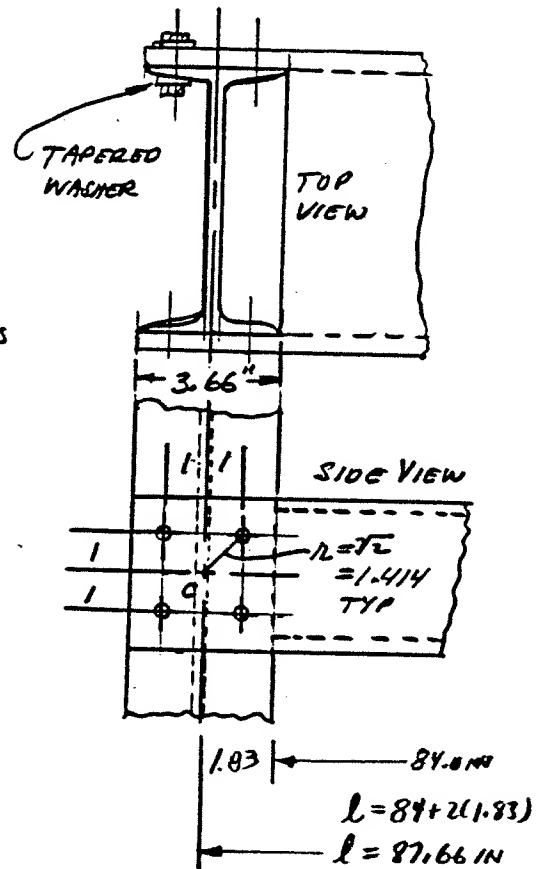
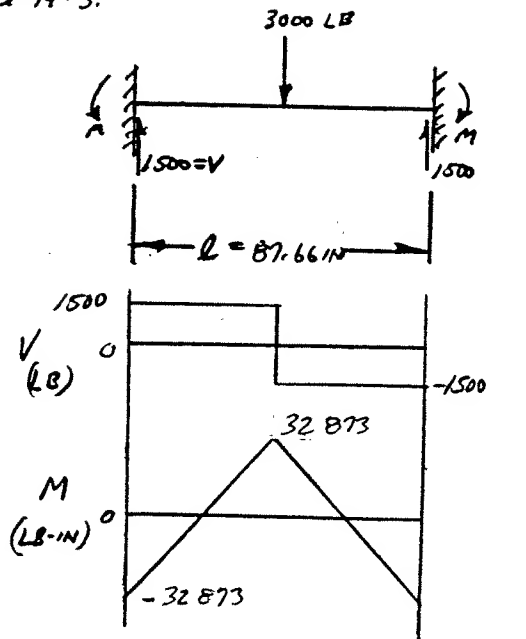
$$\tau_d = 17500 \text{ PSI} = F_R / A_s$$

$$\text{REQ'D. } A_s = F_R / \tau_d = \frac{3041 \text{ LB}}{17500 \text{ LB/IN}^2} = 0.174 \text{ IN}^2$$

$$A_s = \pi D^2 / 4; D = \sqrt{4A_s / \pi} = \sqrt{4(0.174) / \pi}$$

$$D = 0.470 \text{ IN}$$

USE 1/2-13 UNC BOLTS. NO THREADS IN THE SHEAR PLANE.



5.

6 BOLTS, A307, 1400 LB/SIDE

$$\text{SHEAR: } \frac{1400 \text{ LB}}{6} = 233 \text{ LB/BOLT} \downarrow$$

$$M = (1400 \text{ LB})(13 \text{ IN}) = 18200 \text{ LB}\cdot\text{IN}$$

$$r_1 = \sqrt{2^2 + 3^2} = 3.61 \text{ IN}$$

$$r_2 = 2.00 \text{ IN}$$

$$\Sigma r^2 = 4(3.61)^2 + 2(2.00)^2 = 60 \text{ IN}^2$$

$$\text{BOLT } \textcircled{1} \quad F = \frac{M r_1}{\Sigma r^2} = \frac{(18200)(3.61)}{60} = 1095 \text{ LB}$$

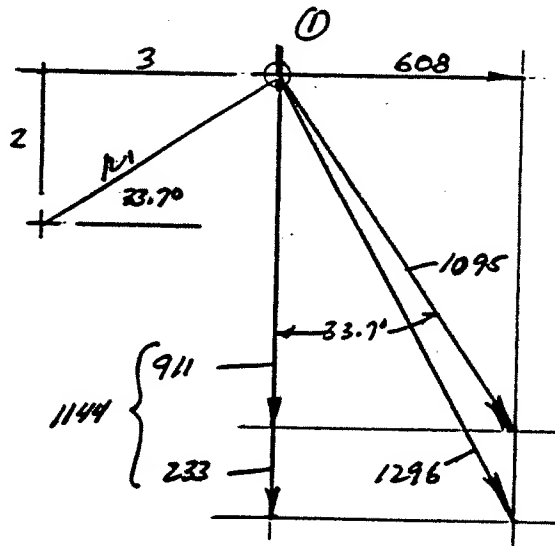
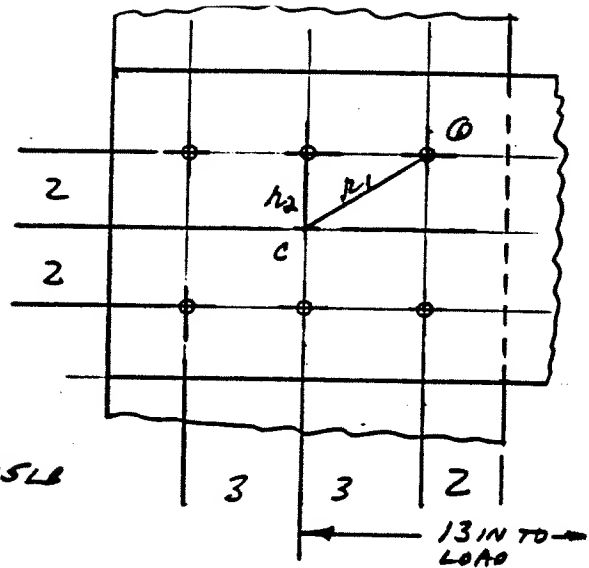
$$\text{TOTAL } F = 1296 \text{ LB}$$

$$A_s = \frac{F}{F_u} = \frac{1296 \text{ LB}}{10000 \text{ LB/IN}^2}$$

$$A_s = 0.130 \text{ IN}^2 = \pi D^2/4$$

$$D = \sqrt{\frac{4 A_s}{\pi}} = \sqrt{\frac{4(0.130)}{\pi}} = 0.406 \text{ IN.}$$

USE 7/16-14 BOLTS



6. FIG. P20-6

VERTICAL COMPONENT OF FORCE
IN EACH CABLE MUST BE 2000 LB

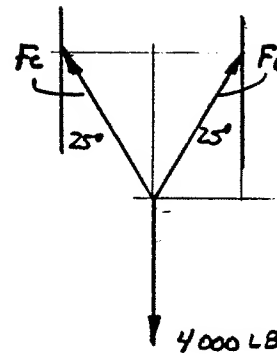
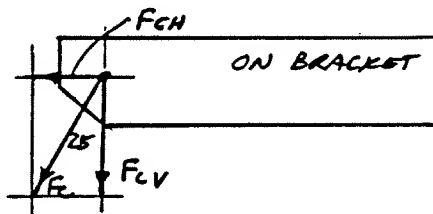
$$F_C \cos 25^\circ = 2000 \text{ LB} = F_{CV}$$

$$F_C = 2000 / \cos 25^\circ = 2207 \text{ LB}$$

$$F_{CH} = F_C \sin 25^\circ$$

$$= 2207 \sin 25^\circ$$

$$F_{CH} = 933 \text{ LB}$$



8 BOLTS, $r = 3.00 \text{ IN}$

A325

$$\Sigma A^2 = 8(3.0)^2 = 72 \text{ IN}^2$$

$$F_A = \frac{M}{\Sigma A^2} = \frac{60000(3)}{72}$$

$$F_A = 2500 \text{ LB}$$

$$\text{SHEAR-HORIZ. } F_1 = 933/8 = 117 \text{ LB} \leftarrow$$

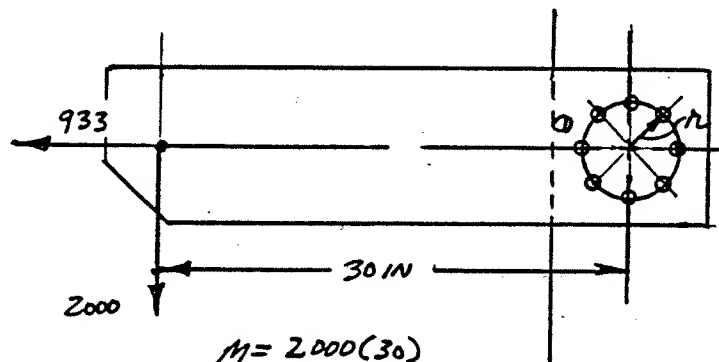
$$\text{SHEAR-VERT. } F_2 = 2000/8 = 250 \text{ LB} \downarrow$$

$$F_T = \sqrt{250^2 + 117^2} = 2753 \text{ LB}$$

$$A_s = \frac{F_T}{T_d} = \frac{2753}{17500} = 0.157 \text{ IN}^2 = \pi D^2/4$$

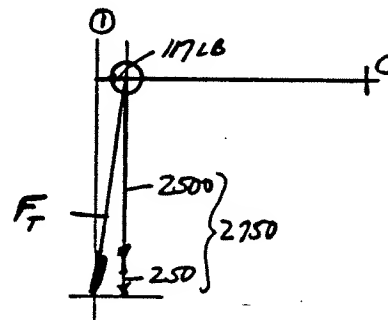
$$D = \sqrt{4A_s/\pi} = \sqrt{4(0.157)/\pi} = 0.448 \text{ IN}$$

USE 1/2-13 BOLTS



$$M = 2000(30)$$

$$= 60000 \text{ LB}\cdot\text{IN}$$



WELDED JOINTS

7.

FIGURE P204

DIRECT SHEAR; $f = V/A_w$

WELD BOTH VERTICAL SIDES, 4.0 IN. LONG. $A_w = 2(4) = 8.0 \text{ IN}$

$$f = \frac{V}{A_w} = \frac{12000 \text{ LB}}{8.0 \text{ IN}} = 1500 \text{ LB/IN}$$

FOR A36 STEEL AND 560 ELECTRODE; $f_u = 9600 \text{ LB/IN/IN}$

$$w = \frac{f}{f_u} = \frac{1500 \text{ LB/IN}}{9600 \text{ LB/IN/IN}} = 0.156 \text{ IN}; \text{ USE } w = \underline{3/16 \text{ IN LEG.}}$$

8.

FROM PROBLEM 4 - AT EACH END OF THE BEAM, $V = 1500 \text{ LB}$
 $T = 32873 \text{ LB}\cdot\text{IN}$ - SHARED EQUALLY ON FRONT AND BACK OF
 PLATES THAT ARE WELDED TO 57×153 COLUMN.

WELD ALONG TOP AND BOTTOM OF PLATE.

CASE 3 - FIG. 20-8: $A_w = 2b = 2(3.66) = 7.32 \text{ IN}$.

$$J_w = (b^3 + 3bd^2)/6 = (3.66^2 + 3(3.66)(4.00)^2)/6$$

$$J_w = 37.45 \text{ IN}^3 \text{ TORSION}$$

ON EACH WELD PATTERN: $V = 750 \text{ LB}$, $T = 16436 \text{ LB}\cdot\text{IN}$.

AT (A): $f_1 = \frac{V}{A_w} = \frac{750 \text{ LB}}{7.32 \text{ IN}} = 102 \text{ LB/IN} \downarrow$

$$f_2 = \frac{T c_y}{J_w} = \frac{(16436 \text{ LB}\cdot\text{IN})(2.0 \text{ IN})}{37.45 \text{ IN}^3} = 878 \text{ LB/IN} \rightarrow$$

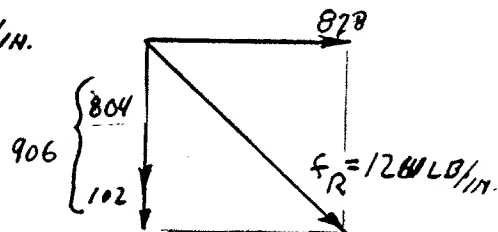
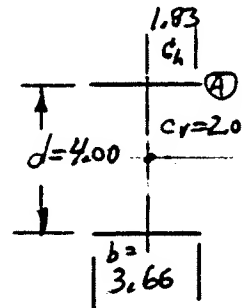
$$f_3 = \frac{T c_x}{J_w} = \frac{(16436)(1.83)}{37.45} = 804 \text{ LB/IN} \downarrow$$

VECTOR SUM OF FORCES: $F_R = 1211 \text{ LB/IN}$.

USE E60 ELECTRODE:

$$w = \frac{F_R}{f_u} = \frac{1211}{9600} = 0.126 \text{ IN}$$

USE $w = \underline{3/16 \text{ IN}} = 0.188 \text{ IN}$ - MINIMUM
 FOR $1/2 \text{ IN PLATE}$.



9.

FIGURE P20-5: TRIM SIDES SO THAT 2.00 IN EXTEND ONTO RIGID STRUCTURE. WELD TOP AND BOTTOM ONLY.

EACH SIDE CARRIES 1400 LB

TORQUE ON WELD: $T = 1400(8 + \frac{b}{2})$

$$T = 1400(8 + 1) = 12600 \text{ LB}\cdot\text{IN}$$

$$A_w = 2b = 2(2) = 4.00 \text{ IN}$$

$$J_w = \frac{b^3 + 3bd^2}{6} = \frac{(2)^3 + 3(2)(8)^2}{6} = 65.3 \text{ IN}^3$$

CASE 3 -
FIGURE 20-8

$$f_1 = \frac{V}{A_w} = \frac{1400 \text{ LB}}{4.00 \text{ IN}} = 350 \text{ LB/IN} \downarrow$$

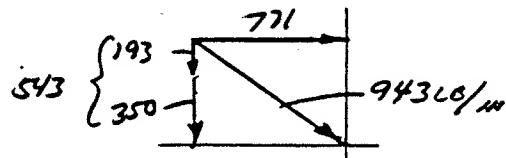
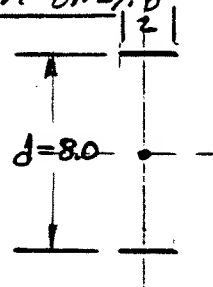
$$f_2 = \frac{T C_v}{J_w} = \frac{(12600 \text{ LB}\cdot\text{IN})(4.00 \text{ IN})}{65.3 \text{ IN}^3} = 771 \text{ LB/IN} \rightarrow$$

$$f_3 = \frac{T C_h}{J_w} = \frac{(12600)(1.00)}{65.3} = 193 \text{ LB/IN} \downarrow$$

E60 ELECTRODES

$$w = \frac{943}{9600} = 0.098 \text{ IN}$$

$$\text{USE } w = \frac{3}{16}'' = 0.188 \text{ IN (MIN)}$$



10.

FIG. P20-6

$$A_w = 2(20) = 20 \text{ in}$$

$$J_u = \frac{d(3b^2 + d^2)}{6}$$

$$J_w = \frac{10(3(10)^2 + 10^2)}{6} = 667 \text{ in}^4$$

AT (A)

$$f_1 = \text{SHEAR} = \frac{933}{20} = 46.7 \text{ LB/in} \rightarrow$$

$$f_2 = \text{SHEAR} = \frac{2000}{20} = 100 \text{ LB/in} \downarrow$$

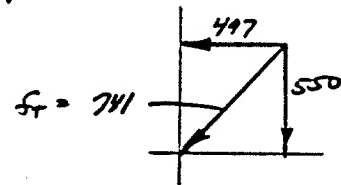
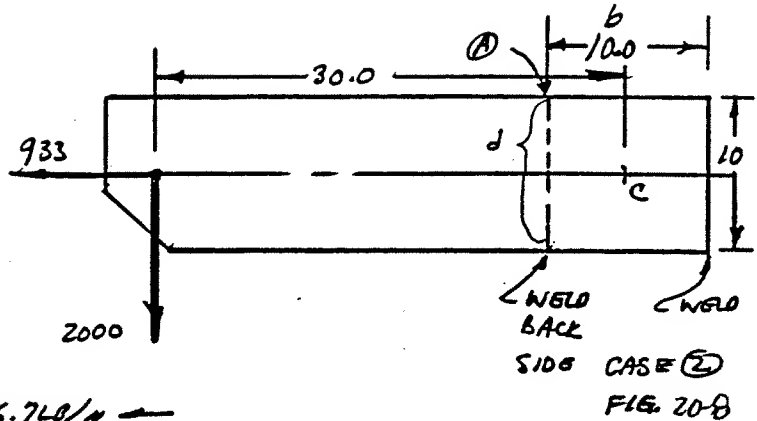
$$f_3 = \text{TORSION} = \frac{T C_v}{J_w} = \frac{2000(30)(5)}{667} = 450 \text{ LB/in} \rightarrow$$

$$f_4 = \text{TORSION} = \frac{T C_1}{J_w} = f_3 = 450 \text{ LB/in} \downarrow$$

$$f_1 + f_3 = 497 \text{ LB/in} \left. \vphantom{\begin{matrix} f_1 + f_3 \\ f_2 + f_4 \end{matrix}} \right\} f_r = 741 \text{ LB/in}$$

$$f_2 + f_4 = 550 \text{ LB/in}$$

$$u = \frac{f_r}{f_w} = \frac{741}{9600} = 0.0771 \text{ in} - \text{USE } u = \frac{3}{16} \text{ (MIN)}$$



11.

FIGURE P20-11

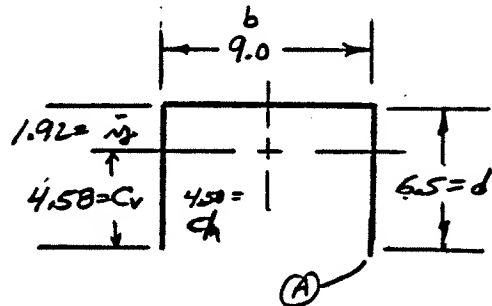
CASE (B), FIG. 20-8

$$A_w = b + 2d = 9 + 2(6.5) = 22.0 \text{ in}$$

$$\bar{y} = \frac{d^2}{b + 2d} = \frac{(6.5)^2}{9 + 2(6.5)} = 1.92 \text{ in}$$

$$I_w = \frac{d^2(2b + d)}{3(b + d)} = \frac{(6.5)^2(2(9) + 6.5)}{3(9 + 6.5)} = 22.3 \text{ in}^4 \text{ (BOTTOM)}$$

$$J_w = \frac{(b + 2d)^3}{12} - \frac{d^2(b + d)^2}{b + d} = 887 - 461 = 426 \text{ in}^4$$



11. (CONTINUED) AT (A):

$$f_1 = \text{SHEAR} = \frac{V}{A_w} = \frac{4200 \text{ LB}}{22.0 \text{ IN}} = 191 \text{ LB/IN} \downarrow$$

$$f_2 = \text{BENDING} = \frac{M}{S_w} = \frac{4200(21.5)}{22.3} = 4056 \text{ LB/IN (INTO WALL)}$$

$$f_3 = \text{TORSION (HORIZ.)} = \frac{T_{CY}}{J_w} = \frac{1200(165)(4.50)}{426} = 213 \text{ LB/IN} \leftarrow$$

$$f_4 = \text{TORSION (VERT)} = \frac{T_{CA}}{J_w} = \frac{1200(165)(4.50)}{426} = 209 \downarrow$$

$$f_1 + f_4 = 191 + 209 = 400 \text{ LB/IN} \downarrow$$

$$\text{RESULTANT} = f_T = \sqrt{f_2^2 + f_3^2 + (f_1 + f_4)^2} = \sqrt{4056^2 + 213^2 + 400^2}$$

$$f_T = 4081 \text{ LB/IN} \quad \text{USE E70 ELECTRODE}$$

$$\text{REQ'D } w = \frac{4081}{11200} = 0.364 \text{ IN} \quad \text{USE } \frac{3}{8} \text{ IN } = w$$

12. FIG P20-11 WITH $P_L = 0$; NO TORSION

$$f_1 = \text{SHEAR} = \frac{V}{A_w} = \frac{3000}{22.0} = 136 \text{ LB/IN} \downarrow$$

$$f_2 = \text{BENDING} = \frac{M}{S_w} = \frac{3000(21.5)}{22.3} = 2892 \text{ LB/IN (INTO WALL)}$$

$$f_T = \sqrt{f_1^2 + f_2^2} = \sqrt{136^2 + 2892^2} = 2896 \text{ LB/IN}$$

USE E60 ELECTRODE:

$$w = \frac{2896}{9600} = 0.302 \text{ IN} \quad \text{USE } \frac{5}{16} \text{ IN, } 0.312 \text{ IN.}$$

13.

FIGURE P20-1 $P = 4000 \text{ LB}$; 6061 ALUMINUM

DIRECT SHEAR - DATA FROM TABLE 19-2

$$\tau = \frac{P}{A_s} = \frac{P}{0.707 w L}$$

 $\sqrt{} = t$ THROAT WIDTH

$$A_s = \text{SHEAR AREA} = (0.707 w) \times \text{LENGTH OF WELD}$$

FOR 4043 FILLER ALLOY, $T_{\text{ALLOW}} = 5000 \text{ PSI}$

$$\text{LET } L = 2(4.00 \text{ IN}) = 8.00 \text{ IN (VERTICAL SIDES OF BRACKET)}$$

$$\text{REQ'D } w = \frac{P}{0.707 L T_{\text{ALLOW}}} = \frac{4000 \text{ LB}}{0.707(8.00 \text{ IN})(5000 \text{ LB/IN}^2)} = 0.141 \text{ IN.}$$

USE $3/16 \text{ IN. WELD (MIN.)}$.

14.

FIGURE P20-14

DIRECT SHEAR, 6061 ALUMINUM

4043 FILLER ALLOY

$$\text{LENGTH OF WELD} = L = \pi D = \pi(4.50) = 14.1 \text{ IN}$$

$$\text{AS IN 13: } w = \frac{P}{0.707 L T_n} = \frac{1500}{(0.707)(14.1)(5000)} = 0.030 \text{ IN.}$$

USE $3/16 \text{ IN}$

COULD ALSO CONSIDER PARTIAL WELD.

15.

FIGURE P20-15

DIRECT SHEAR, 6063 ALUM.

4043 FILLER

$$\text{LET } w = 3/16 = 0.188 \text{ IN}$$

SOLVE FOR L

$$L = \frac{P}{0.707 w T_{\text{ALLOW}}} = \frac{800 \text{ LB}}{(0.707)(0.188)(5000 \text{ LB/IN}^2)} = 1.20 \text{ IN.}$$

DISTRIBUTE PARTIAL WELDS TOTALING 1.20 IN
EQUALLY ON BOTH SIDES OF TAB.

16. FIGURE P20-16 DIRECT SHEAR, 5003 ALUMINUM
4043 FILLER ALLOY

AS IN PROBLEM 14:

$$\text{REQ'D. } W = \frac{P}{0.707 L T_u}$$

$$T_u = 5000 \text{ PSI}$$

$$L = 2(\pi D) = 2\pi(2.0 \text{ IN}) = 12.57 \text{ IN. WELD ALL AROUND}$$

$$W = \frac{10000 \text{ LB}}{(0.707)(12.57 \text{ IN})(5000 \text{ LB/IN}^2)} = 0.225 \text{ IN}$$

$$\text{USE } W = 1/4 \text{ IN} = 0.250 \text{ IN}$$

17. MATERIAL COMPARISON

$$F = 4800 \text{ LB}; N = 2; \sigma_b = S_y/2$$

$$\text{REQ'D } A = F/\sigma_b; D = \sqrt{4A/\pi}$$

COMPUTE WEIGHT PER INCH OF LENGTH

$$V = A \times 1.0 \text{ IN}$$

$$W = \rho_w V$$

MATERIAL	$S_y(\text{KSI})$	$\sigma_b(\text{KSI})$	$A(\text{IN}^2)$	$D(\text{IN})$	$V(\text{IN}^3)$	$\rho_w(\text{LB/IN}^3)$	$W(\text{LB})$
a) 1020 HR	30	15	.32	.638	.32	.283	.0906
b) S160 OQT1300	100	50	.096	.350	.096	.283	.0272
c) ALUM. 2014-T6	60	30	.16	.451	.16	.100	.0160
d) ALUM 7075-T6	73	36.5	.132	.409	.132	.100	.0132
e) Ti-6AL-4V (ANNEALED)	120	60	.080	.319	.080	.160	.0128
f) Ti-3AL-3V-11CR	175	87.5	.055	.264	.055	.160	.0088

CHAPTER 21 ELECTRIC MOTORS AND CONTROLS

Questions 1 - 8: See Sections 21-2 and 21-3.

9. Standard frequency for AC power in the U.S. is 60 hertz.
10. Standard frequency for AC power in Europe is 50 hertz.
11. Single phase AC power at 115 and 230 volts.
12. Two conductors plus a ground wire.
13. 480V, three phase is preferred because the current would be lower and the size of the motor would be smaller.
14. Synchronous speed is the speed at which an AC motor tends to run at zero load. $n_s = 120(f)/p$, where f is the frequency of the power and p is the number of poles in the motor.
15. Full-load speed is the speed of the motor when it is delivering its rated torque.
16. In U.S.: $n_s = 120(f)/p = 120(60)/4 = 1800$ rpm
In France: $n_s = 120(f)/p = 120(50)/4 = 1500$ rpm
17. 2-pole motor. Zero-load speed approximately 3600 rpm.
18. $n_s = 120(f)/p = 120(400)/4 = 12000$ rpm
19. Two speed motor; 1725 rpm and 1140 rpm
20. Variable frequency control
- 21,22,24 - See Section 21-8.
23. National Electrical Manufacturers Association
25. TEFC - Totally enclosed - fan-cooled. See Section 21-8.
26. TENV - Totally enclosed - non-ventilated. Section 21-8.
27. NEMA Design 9 - Hazardous locations. Flour can explode.
28. TENV because motor may get bathed in water during cleaning and to protect food from contaminants from the motor.

30. Locked rotor torque is the torque that a motor can exert when the rotor is at rest. Also called starting torque.
31. A poorer speed regulation means that the motor would slow down more when subjected to an increase in torque.
32. Breakdown torque is the maximum torque a motor can develop during the increase in speed after start or the torque at which a motor would be stalled if the torque is increased after it is running.
33. Split-phase; capacitor-start; permanent-split capacitor; shaded pole.
34. a) Single phase, split-phase AC motor because of the moderate starting torque and the change of torque when the switch cuts out the starting winding.
 b) From Table 21-2, full-load speed = 1140 rpm
 $T = 63000(P)/n = 63000(.75)/1140 = 41.4 \text{ lb-in}$
 c) Starting torque = 150%(F.L. torque)
 $T_s = 1.5(41.4) = 62.2 \text{ lb-in (approximate)}$
 d) Breakdown = 350%(F.L. torque)
 $T_b = 3.5(41.4) = 145 \text{ lb-in (approximate)}$
35. 2-pole; 1.50 kW rated power.
 b) From Table 21-2, full-load speed = 3450 rpm
 $n = (3450 \text{ rev/min}) (2\pi \text{ rad/rev}) (1 \text{ min}/60 \text{ s}) = 361 \text{ rad/s}$
 $T = P/n = (1.5 \times 10^3 \text{ N-m/s}) / (361 \text{ rad/s}) = 4.15 \text{ N-m}$
 c) Starting torque = 1.5(4.15 N-m) = 6.23 N-m
 d) Breakdown torque = 3.5(4.15 N-m) = 14.5 N-m
36. Fan requires about 18 lb-in of torque at 1725 rpm; low starting torque; assume fan cools motor. Recommend 4-pole, single phase permanent split capacitor AC motor.
 $\text{Power} = Tn/63000 = (18)(1725)/63000 = 0.49 \text{ hp (use } 1/2 \text{ hp)}$
37. Full load torque about 0.5 N-m at 3450 rpm; high starting torque (about 2.8xF.L.T.) due to starting compressor against high pressure in the system. Recommend capacitor start, single phase, 2-pole, AC motor.
 $n = (3450 \text{ rev/min}) (2\pi \text{ rad/rev}) (1 \text{ min}/60 \text{ s}) = 361 \text{ rad/s}$
 $P = Tn = (0.5 \text{ N-m}) (361 \text{ rad/s}) = 181 \text{ N-m/s} = 181 \text{ watts}$
38. Speed is adjusted by varying the resistance in the rotor circuit through an external resistance control.
39. F.L. speed = synchronous speed = 720 rpm. (Table 21-2)

40. Pull-out torque is the torque that would disengage the motor from its synchronous speed and cause it to stop.
41. Universal motors are very small and light weight for a given power rating. Some vacuums, appliances, and hand tools utilize the high speed of rotation effectively.
42. A universal motor can operate on DC or almost any frequency of AC voltage when operating near its full-load point.
43. Batteries, generators, rectified AC, *HYDROGEN FUEL CELLS*.
44. See Table 21-7.
45. SCR - Silicon controlled rectifier. Used to produce DC power from AC.
46. SCR controls do not produce pure DC power; it has some variation, called ripple, due to the AC input. A low-ripple control would produce a nearly true DC power.
47. Could use a 90V DC motor powered from a NEMA Type K SCR power supply to convert 115 V AC to 90V DC power.
- 48-50. See Section 21-11.
51. The motor would speed up without limit and may fail catastrophically.
52. Speed is proportional to torque. $T_2 = T_1 (n_1/n_2)$
 $T_2 = (15.0 \text{ N-m}) (3000/2200) = 20.5 \text{ N-m}$
- 53, 56-61. See Sections 21-9, 21-11, and 21-12.
54. NEMA Size 2 motor starter for 10 hp, 220V AC, 3-phase.
55. NEMA Size 1 motor starter for 1.0 kW, 110V AC, single phase.

CHAPTER 22

MOTION CONTROL: CLUTCHES AND BRAKES

1. FROM EQ. 22-1: $T = C P K / m = (63025)(5.0)(2.75) / 1750 = 495 \text{ LB} \cdot \text{IN}$
 C AND K FROM SECTION 22-4.

SUMMARY OF RESULTS FOR PROBLEMS 2-7.

PROB.	C	P	K	m	TORQUE
2	5252	75 hp	5.0	2500	788 LB·FT
3	63025	0.50 hp	1.5	1150	41 LB·IN
4	63025	5.0 hp	2.75	180	4814 LB·IN
5-1	63025	5.0 hp	1.0	1750	180 LB·IN
5-2	5252	75 hp	1.0	2500	158 LB·FT
5-3	63025	0.50 hp	1.0	1150	27.4 LB·IN
5-4	63025	5.0 hp	1.0	180	1751 LB·IN
6	9549	20 kW	2.75	3450	152 N·m
7(a)	9549	50 kW	4.0	900	2122 N·m CLUTCH
(b)	9549	50 kW	1.0	900	531 N·m BRAKE

8. DISK: $D = 24.0 \text{ IN}$; $R_1 = 12.0 \text{ IN}$; $L = 2.50 \text{ IN}$; $R_2 = 0$; STEEL
- $$Wk^2 = \frac{L(R_1^4 - R_2^4)}{323.9} = \frac{2.50(12.0^4 - 0)}{323.9} \text{ LB} \cdot \text{FT}^2 = 160.0 \text{ LB} \cdot \text{FT}^2$$

$$T = \frac{Wk^2(\Delta \omega)}{308t} = \frac{(160)(550)}{308(2.0)} = 142.9 \text{ LB} \cdot \text{FT}$$

- 9.
- | | R_1 | R_2 | L | Wk^2 |
|----------|-------|-------|------|---------------------------|
| SHAFT | 0.625 | 0 | 16.0 | 0.00754 |
| COUPLING | 1.50 | .625 | 2.25 | 0.0341 |
| BRG-1 | 1.00 | .625 | 1.80 | 0.0047 |
| HUB | 2.00 | .625 | 1.00 | 0.0489 |
| GEAR | 6.00 | .625 | 3.00 | 12.0023 |
| BRG-2 | 1.00 | .625 | 1.80 | 0.0047 |
| TOTAL | | | | 12.102 LB·FT ² |
- $$T = \frac{Wk^2(\Delta \omega)}{308t} = \frac{(12.102)(775)}{308(1.50)} = 60.9 \text{ LB} \cdot \text{FT}$$

10.

NEGLECT CLUTCH AND SHORT SHAFT BETWEEN CLUTCH AND GEAR A.

SPEED OF SHAFT 2: $M_2 = 1750 (400/1500) = 466.7 \text{ RPM}$

IN EQ. 22-4 $(M/M_2)^2 = (M_2/M_1)^2 = (466.7/1750)^2 = 0.0711$

	<u>R₁</u>	<u>R₂</u>	<u>L</u>	<u>WR²</u>	<u>WR²_e</u>
GEAR A	2.00	0	3.00	0.1482	0.1482
GEAR B	7.50	1.25	3.00	29.2833	2.0824
SHAFT 2	1.25	0	54.0	0.4070	0.0289
HUB 1	2.50	1.25	5.0	0.5653	0.0402
HUB 2	2.50	1.25	5.0	0.5653	0.0402
END PLATES	9.00	1.25	2.00	40.4974	2.8798
HOLLOW CYL	9.00	7.50	30.0	314.6284	22.3736
			TOTAL		27.5933 LB-FT ²

$$T = \frac{WR_e^2(\text{lb-in})}{308 \text{ ft}} = \frac{(27.5933)(1750)}{308(1.50)} = 104.5 \text{ LB-FT}$$

11.

LOAD SPEED = 50 FT/MIN = V

$$\text{DRUM SPEED} = \frac{V}{R} = \frac{50 \text{ FT}}{\text{MIN}} \times \frac{1}{4.10 \text{ IN}} \times \frac{12 \text{ IN}}{\text{FT}} = 150 \text{ RAD/MIN} = \omega$$

$$\text{RPM OF DRUM} = M = \frac{150 \text{ RAD}}{\text{MIN}} \times \frac{60 \text{ S}}{2\pi \text{ RAD}} = 23.87 \text{ RPM}$$

	<u>R₁</u>	<u>R₂</u>	<u>L</u>	<u>WR²</u>
SHAFT	0.75	0	24.0	0.0234
END PLATES	4.00	.75	3.0	2.3682
HOLLOW CYL.	4.00	3.00	13.0	7.0238
			TOTAL	9.4154 LB-FT ²

$$\text{LOAD: } WR_e^2 = W \left(\frac{V}{\omega} \right)^2 = 600 \text{ LB} \left(\frac{50 \text{ FT/MIN}}{150 \text{ RAD/MIN}} \right)^2 = 66.6667 \text{ LB-FT}^2$$

$$\text{TOTAL } WR^2 = 9.4154 + 66.6667 = 76.08 \text{ LB-FT}^2$$

$$\text{BRAKING TORQUE} = \frac{WR^2(\text{lb-in})}{308 \text{ ft}} = \frac{(76.08)(23.87)}{308(0.25)} = 23.6 \text{ LB-FT}$$

$$\text{ADDITIONAL TORQUE TO HOLD LOAD} = 600 \text{ LB}(4 \text{ IN}) \left(\frac{12 \text{ IN}}{\text{FT}} \right) = 200 \text{ LB-FT}$$

$$\text{TOTAL BRAKE TORQUE} = 223.6 \text{ LB-FT}$$

12. (a) CLUTCH ON MOTOR SHAFT:

MOTOR SPEED = CLUTCH SPEED = 1150 RPM = m_c

BARREL SPEED = 38 RPM = m

$$(m/m_c)^2 = (38/1150)^2 = 0.001092$$

	<u>R_1</u>	<u>R_2</u>	<u>L</u>	<u>Wk^2</u>	<u>Wk^2_e</u>
WORM	1.75	0	8.00	0.2316	0.2316
WORM GEAR	8.00	0	2.50	31.6147	0.0344
2 HUBS	4.00	2.00	4.00	2.9639	0.0032
END PLATES	14.00	2.00	4.00	474.2204	0.5178
HOLLOW CYL.	14.00	12.00	18.00	982.5255	1.0728
SHAFT	2.00	0	36.0	1.7783	0.0019
					<u>1.8617 LB-FT²</u>

$$T = \frac{Wk^2_e (\Delta m)}{308 t} = \frac{(1.8617)(150)}{308(2.0)} = \underline{\underline{3.48 \text{ LB-FT}}}$$

(b) CLUTCH ON WORM GEAR SHAFT. ONLY BARREL AND HUBS ARE ACCELERATED TO 38 RPM = CLUTCH SPEED.

	<u>Wk^2</u>	<u>$T = \frac{Wk^2 (\Delta m)}{308 t}$</u>
2 HUBS	2.9639	
END PLATES	474.2204	
HOLLOW CYL.	982.5255	
SHAFT	1.7783	
TOTAL	<u>1461.5 LB-FT²</u>	<u>$T = \frac{(1461.5)(38)}{308(2.0)} = 90.2 \text{ LB-FT.}$</u>

13.

$$T_f = F N R_m ; R_m = T_f / F N = (75 \text{ LB-IN}) / (1.25 \times 150) \text{ LB} = \underline{\underline{2.00 \text{ IN}}}$$

$$P_f = \frac{T_f m}{63000} \quad hp = \frac{(75)(1150)}{63000} = \underline{\underline{1.37 \text{ hp}}}$$

$$\text{LET } WR = 0.10 \text{ hp/in}^2 = P_f / A$$

$$A = P_f / WR = 1.37 \text{ hp} / 0.10 \text{ hp/in}^2 = \underline{\underline{13.7 \text{ in}^2}}$$

$$A = \pi(R_o^2 - R_i^2), R_m = (R_o + R_i)/2 \text{ OR } 2R_m = R_o + R_i$$

$$\text{TRY } R_o = 1.5 R_i ; 2R_m = 1.5 R_i + R_i = 2.5 R_i ; R_i = \frac{2R_m}{2.5} = \frac{2.00}{1.25} = 1.60 \text{ in}$$

$$R_o = 1.5(1.60) = 2.40 \text{ in} ; A = \pi(2.40^2 - 1.60^2) = \underline{\underline{12.05 \text{ in}^2}} \quad \text{LOW}$$

SIMILARLY, FOR $R_o = 1.75 R_i$

$$R_i = 1.45 \text{ in} ; R_o = 2.55 \text{ in} ; A = 13.75 \text{ in}^2 \text{ OK}$$

14. FROM PROB. 9: $T_f = 64 \text{ LB} \cdot \text{IN}$; TRY $R_m = 1.50 \text{ IN}$; $f = 0.25$

$$N = \frac{T_f}{f R_m} = \frac{64.0}{(0.25)(1.50)} = 170 \text{ LB}$$

$$P_f = \frac{T_f N}{63000} = \frac{(64)(170)}{63000} = 0.79 \text{ HP}$$

$$\text{FOR } W_R = 0.10 \text{ HP}/\text{in}^2; A = P_f / W_R = 7.9 \text{ in}^2$$

$$\text{TRY } R_o = 1.50 R_i; R_m = (R_o + R_i) / 2$$

$$2R_m = 1.5R_i + R_i = 2.5R_i$$

$$R_i = R_m / 1.25 = 1.50 \text{ in} / 1.25 = 1.20 \text{ IN}; R_o = 1.5(1.2) = 1.80 \text{ in}$$

$$A = \pi(R_o^2 - R_i^2) = 5.65 \text{ IN}^2 \text{ (LOW)}$$

$$\text{TRY } R_o = 2.0 R_i; 2R_m = 2.0R_i + R_i = 3.0R_i; R_i = R_m / 1.5 = 1.0 \text{ in}$$

$$R_o = 2.0(1.0) = 2.0 \text{ in}; A = \pi(2.0^2 - 1.0^2) = 9.42 \text{ in}^2 \text{ OK}$$

15. FROM EQ. 12-13:

$$F_a = \frac{T_f (\sin \alpha + f \cos \alpha)}{f R_m} = \frac{15 \text{ lb} \cdot \text{ft} (\sin 12^\circ + 0.25 \cos 12^\circ)}{(0.25)(3.0 \text{ in})} \times \frac{12 \text{ in}}{\text{ft}}$$

$$F_a = 109 \text{ lb}$$

16. $T_f = 64 \text{ lb} \cdot \text{in}$; Let $f = 0.25$; $R_m = 2.0 \text{ in}$; $\alpha = 12^\circ$

$$F_a = \frac{(64)(\sin 12^\circ + 0.25 \cos 12^\circ)}{(0.25)(1.0)} = 116 \text{ lb}$$

17. $T_f = 150 \text{ lb} \cdot \text{ft} \times 12 \text{ in}/\text{ft} = 1800 \text{ lb} \cdot \text{in} = F_f D_o / 2$

$$F_f = 2T_f / D_o = 2(1800) / 12.0 = 300 \text{ lb}$$

$$W = \frac{F_f \left(\frac{a}{f} - b \right)}{L} = \frac{(300 \text{ lb}) \left(\frac{4.0}{0.25} - 5.0 \right) \text{ in}}{24.0 \text{ in}} = 138 \text{ lb}$$

18. FOR SELF ACTIVATION; $W < 0$.

$$\text{FOR } W = 0, a/f - b = 0 \text{ OR } b = a/f = 4.0 \text{ in} / 0.25 = 16.0 \text{ IN}$$

$$b > 16.0 \text{ IN FOR SELF ACTIVATION}$$

19. $T_f = 100 \text{ LB} \cdot \text{FT} \times 12 \text{ IN/FT} = 1200 \text{ LB} \cdot \text{IN}$
 TRY $D_o = 10.0 \text{ IN}$; $f = 0.25$

$$F_f = \frac{2T_f}{D_o} = \frac{2(1200)}{10} = 240 \text{ LB}$$

IN FIG 22-17 (C); LET $a = 2.0 \text{ IN}$; $b = 6.00 \text{ IN}$; $f = 0.25$; $L = 18 \text{ IN}$

$$W = \frac{F_f \left(\frac{a}{f} - b \right)}{L} = \frac{240 \left(\frac{2.0}{0.25} - 6.0 \right)}{18} = 26.7 \text{ LB}$$

20. $T_f = 100 \text{ LB} \cdot \text{FT} \times 12 \text{ IN/FT} = 1200 \text{ LB} \cdot \text{IN}$
 SELECT WOVEN ASBESTOS; $f = 0.25$; $p \approx 30 \text{ psi}$
 IN FIG 22-19: $r = 6.0 \text{ IN}$; $C = 9.00 \text{ IN}$; $L = 20.0 \text{ IN}$
 $\theta_1 = 45^\circ$; $\theta_2 = 135^\circ$
 FROM EQ. 22-18:

$$W = \frac{T_f}{r^2 f p (\cos \theta_1 - \cos \theta_2)} = \frac{1200 \text{ LB} \cdot \text{IN}}{(6.0)^2 (0.25) (30) (\cos 45^\circ - \cos 135^\circ)} = 3.25 \text{ IN}$$

USE $W = 3.25 \text{ IN}$; $P = 30 \text{ psi} \left(\frac{3.14}{3.25} \right) = 29.0 \text{ psi}$

$$\theta_2 - \theta_1 = 135^\circ - 45^\circ = 90^\circ \times \frac{\pi}{180} = \pi/2 = 1.57 \text{ rad.}$$

(EQ. 22-20)

$$M_N = 0.25 (29.0) (3.25) (6.0) (9.0) [2(1.57) - \sin 270^\circ + \sin 90^\circ]$$

$$M_N = 6540 \text{ lb} \cdot \text{IN}$$

(EQ. 22-21)

$$M_f = -(0.25) (29.0) (3.25) (6.0) [6.0 (\cos 45^\circ - \cos 135^\circ) + (0.25) (9.0) (\cos 270^\circ - \cos 90^\circ)]$$

$$M_f = -1200 \text{ lb} \cdot \text{IN}$$

(EQ. 22-19)

$$W = (M_N - M_f) / L = (6540 - 1200) / 20 = 267 \text{ LB} = \text{ACTIVATION FORCE}$$

CHECK WEAR RATIO

$P_f = T_f M / 63000 = 1200 (480) / 63000 = 9.14 \text{ hp}$
 (EQ. 22-23)

$$A = 2 W r \sin \left(\frac{\theta_2 - \theta_1}{2} \right) = 2 (3.25) (6.0) \sin 45^\circ = 27.6 \text{ in}^2$$

$$WR = \frac{P_f}{A} = \frac{9.14 \text{ hp}}{27.6 \text{ in}^2} = 0.33 \text{ hp/in}^2 \text{ SOMEWHAT HIGH - INTERMITTENT SERVICE ONLY.}$$

21. BAND BRAKE: $T_f = 75 \text{ LB-FT} (12 \text{ in/FT}) = 900 \text{ LB-IN}$; $n = 350 \text{ RPM}$
 USE WOVEN ASBESTOS, $p_{\text{MAX}} = 25.0 \text{ psi}$; $f = 0.25$
 TRY $r = 6.0 \text{ IN}$; $\theta = 210^\circ (3.67 \text{ RAD})$; $w = 2.5 \text{ IN}$

$$P_1 = p_{\text{MAX}} r w = (25.0)(6.0)(2.5) = 375 \text{ LB}$$

$$P_2 = \frac{P_1}{e^{f\theta}} = \frac{375 \text{ LB}}{e^{(0.25)(3.67)}} = 150 \text{ LB}$$

$$T_f = (P_1 - P_2)r = (375 - 150)(6.0) = 1350 \text{ LB-IN (HIGH)}$$

TRY $r = 5.50 \text{ IN}$; $\theta = 210^\circ$; $w = 2.0 \text{ IN}$

$$P_1 = (25.0)(5.50)(2.0) = 275 \text{ LB}$$

$$P_2 = \frac{275}{e^{(0.25)(3.67)}} = 110 \text{ LB}$$

$$T_f = (275 - 110)(5.5) = 908 \text{ LB-IN OK}$$

FOR SIMPLE BAND BRAKE: LET $a = 5.50 \text{ IN}$; $L = 12.0 \text{ IN}$

$$w = P_2 (a/L) = 110(5.5/12) = 50.4 \text{ LB}$$

WEAR RATIO:

$$A = 2\pi r w \frac{\theta}{360} = 2\pi(5.5)(2.0) \frac{210}{360} = 40.3 \text{ in}^2$$

$$P_f = \frac{T_f n}{63000} = \frac{(908)(350)}{63000} = 5.04 \text{ HP}$$

$$WR = P_f/A = 5.04/40.3 = 0.125 \text{ HP/in}^2 \text{ OK}$$